STEAM POWER AND MILL WORK

THE SPECIALISTS' SERIES

STEAM POWER AND MILL WORK

PRINCIPLES AND MODERN PRACTICE

BY

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PREFACE

This work aims at giving an account of modern practice for the consideration of those interested in the manufacture, control, and operation of boilers, engines, and millwork. Also of the leading principles and calculations affecting such work. Most of the subject matter of the work is based upon the personal experience of the author.

Special attention has been paid to simplicity of treatment of mathematical questions, in the belief that this course will meet with the approval of the majority of readers, and that the real utility of the work will be thereby increased.

The author believes that the work will be found to be of use in deciding questions as to the provision of new plant, or the conversion or replacement of existing plant, and in determining the conditions under which it may be worked to best advantage.

In all extensive manufacturing operations, the provision of power is necessary; and in many cases the success of the work depends upon the efficiency of the means adopted for this purpose. In every direction evidence is found to show that no district or nation is by nature secure against loss of business. Such security is only to be assured by making the best use of every natural advantage, and surmounting every difficulty or

disadvantage in the best way. Probably the British nation enjoys unequalled natural advantages, but others are evincing great aptitude in compensating for all disadvantages by the adoption of improved practice.

By the adoption of the manifold expansion of high-pressure steam, the use of superheated steam, the complete trials of boilers and engines, with detailed analyses of results, the analysis of furnace gases, calorimetric testing of fuels, superior cutting and finishing of wheel-teeth, and many other measures, important advantages are secured, which, though well known in England, and in the abstract approved, are frequently considered to be unnecessary refinements.

In the preparation of the work, the author has gratefully received most valuable assistance from a large number of friends. Useful information has also been obtained from the *Minutes of Proceedings* of the Institution of Civil Engineers, the Institution of Mechanical Engineers, and the Institution of Naval Architects, by permission of their respective Councils; also from various technical journals.

A few brief repetitions may be noticed, chiefly upon very important points. Electric transmission of power, which is very much discussed at the present time, is not dealt with in this work, but in that of Mr. Kapp in the same series.

G. W. SUTCLIFFE.

Marple, near Stockport,
December 1894.

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CHAPTER I.

HEAT AND WORK.

Source of heat.—According to the existing natural constitution of the earth, the heat of the sun is the practical source of all known forms of useful energy. Steam and other heat engines, also vital force, and the forces depending upon falling water, the winds, and electrical and chemical separation, all depend upon heat, and would rapidly become extinct if the heat of the sun were withdrawn. The central heat of the earth gives rise to force which is manifested in earthquakes, volcanic disturbances, geysers, &c. The power of these has, however, not been hitherto applied to any useful purpose, and in all probability it also is originally connected with the heat of the sun.

Nature and effects of heat.—Heat is a condition of matter, and is not known to exist apart from matter. A quantity of matter—say one pound—weighs exactly the same whether hot or cold, but its volume, density, colour, chemical affinities and other properties vary

with its temperature. At sufficiently low temperatures, all bodies assume the solid form. On the other hand, at sufficiently high temperatures solid matter becomes liquid, and at still higher temperatures assumes the form of vapour or gas. Professor Dewar has lately caused air and other gases to assume liquid form and alcohol the solid form; in each case the result had been commonly predicted, but the means whereby exceedingly low temperatures have been secured are novel, and display the highest degree of manipulative skill. These results are of great importance, as confirming the probability of ultimate success in the liquefaction and freezing of the few substances which have not already yielded; also in showing the marvellous inactivity of chemical affinity at low temperatures, and affording further evidence of the dependence of all sources of energy upon presence of heat.

Quantity of heat imparted to the earth by the sun.—It has been calculated that the sun pours upon the earth's atmosphere in one year a quantity of heat sufficient to melt a coating of ice one hundred feet thick over the whole surface, but that one-half of this is intercepted by the atmosphere before reaching the solid earth. The remaining half would suffice to furnish 780 horse-power per acre night and day for the whole year if applied without loss, or say 100 horse-power as applied in an ordinary steam engine. There is no immediate probability that this source of heat can be utilized under British skies, but the observations of Ericsson and others tend to show that the heat poured upon tropical deserts may be utilized at some time, and perhaps transmitted electrically to great distances.

Heat of sun is indirectly utilized.—But though the rays of the sun cannot yet be collected and utilized directly, this can be done through the agency of vital

force, which is directly dependent upon the sun. In this way food is furnished for men and animals whereby they are enabled to perform work. The same food could be burnt for the generation of steam to be utilized in an engine, or the food may be consumed in other ways. But the amount of work thus produced by its means would be less than in the first case, on account of the amount of waste being greater than in the animal economy. Some kinds of vegetable matter—as, for instance, wood—are not in their nature suitable for the support of animal life, and therefore they can only be utilized for the production of energy or work by means of burning. A very large proportion of the mechanical energy utilized in the world is produced in the combustion of vegetable matter arising from plants which flourished and died in past ages, and which has undergone changes, which have led to the production of coal. Peat and lignite are of vegetable origin, but less advanced in change than is coal. The above substances and also natural gas, mineral oils and substances produced from any of them, are burnt to produce heat, with a view to the ultimate production of energy or work. Combustion or burning depends upon the chemical affinity which exists between the chief constituents of the fuel and the oxygen of the atmosphere. In this combination heat is produced, the amount of which exactly corresponds to the heat absorbed in the process of growth, and which heat was originally derived from the rays of the sun. A suitable medium is required, to which the heat is imparted for effecting its conversion into work, for which purpose water is usually but not invariably adopted.

Power of falling water, wind, and tides.—The power of falling water is also derived from the heat of the sun, by whose action the atmosphere is caused to

imbibe water vapour from every exposed surface of water and from moist substances. This vapour is carried along by winds and currents, and is deposited as rain, the amount of which is greatest in hilly countries, and at one particular altitude in each district. The rainfall in the British Islands rarely falls below thirty inches. If only ten inches of this can be utilized for the purpose of producing power, a watershed area of twelve square miles will produce an average of one horse-power per foot of fall throughout the year, day and night without stoppage. In districts where chalk or other open formations prevail, the amount of water to be obtained is very uncertain. But in hilly districts, upon formations free from limestone, and not seriously disturbed by mining operations, five square miles will usually produce water power to the amount of one horse-power per foot of fall, during ordinary working hours with little storage. This obviously cannot be obtained in the dryest weather. The situations in which water power is abundant are not infrequently placed at a great disadvantage in respect to communication and transit of goods; and in the immediate neighbourhood of seaport towns, which possess the greatest advantages as to transit, good water power is very seldom to be obtained. Winds and tides are very powerful in the aggregate. But their power is too diffusive for economical application. They are also often placed at a disadvantage by reason of difficulty of access to the localities in which their power is most largely exhibited. For instance, winds are most powerful and constant upon the top of a hill, which is generally a most inconvenient situation.

Temperature.—The temperature of a body measures its condition with respect to sensible heat. The body will receive heat from one of a higher temperature,

and will impart heat to one of lower temperature. Temperature is expressed in degrees. The scale adopted in Great Britain is that arranged by Fahrenheit in 1714. He used a freezing mixture of snow and salt, which he believed to be absolutely destitute of heat, and adopted its temperature as 0° on the scale. He then divided the interval between 0° and that of freezing water (or melting ice) into 32 degrees, which, being continued upwards, reached the temperature of boiling water under mean atmospheric pressure at 212°, or 180° above freezing point. Fahrenheit's idea was an excellent one, but unfortunately he had no means whereby to predict the absolute zero of temperature as estimated at the present time. Celsius soon afterwards proposed the centigrade scale, in which the zero is placed at the freezing point of water, and 100° at the boiling point of water under mean atmospheric pressure. This scale possesses points of greater convenience than Fahrenheit's, but hitherto they have not been considered of importance sufficient to secure general acceptance in Great Britain.

Absolute zero of temperature, or extinction of heat.—In observations upon the expansion of gases by heat and in other ways, good evidence is obtained to show that at −460° F., or 492° below the freezing point of water (−273° C.), all heat would disappear; this point is therefore referred to as absolute zero. The recent experiments of Professor Dewar become even more interesting when considered with reference to this temperature.

Measurement of heat.—Quantities of heat are estimated in units. A British thermal unit is the amount of heat which will raise one pound of water from a temperature of 39° F. to 40°. The total amount required to raise the same quantity of water from 32° F. to 212°

F. is 180·9 units, or one-half per cent. greater than if constant.

Effects of application of heat.—When heat is imparted to a body, it may be expended solely in increasing the sensible heat of the body as measured by temperature. Such increase will be approximately in proportion to the quantity of heat imparted, and inversely in proportion to the mass or quantity of material receiving heat. Heat may also be expended in causing a change in the condition of the matter to which it is imparted. A solid body may be changed into a liquid absorbing an amount of heat, termed the "latent heat of liquefaction"; and a liquid may be changed into a gas or vapour, absorbing in the process "latent heat of vaporization." Latent heat is also absorbed when a soluble substance assumes liquid form by solution in water or other solvent fluid. When such an operation is effected without the supply of heat from outer sources, the temperature falls. In each case the latent heat is constant in amount for any given substance, under given conditions, but varies with pressure. The separation of solid matter from solution liberates heat to an equal extent with that absorbed in the process of solution. The latent heats of vaporization and liquefaction are also precisely restored in the reverse processes. Heat is absorbed and temperature reduced when water is evaporated by the action of the atmosphere, and is liberated when the water falls as rain. In the occurrence of snow, the combined latent heats of vaporization and liquefaction are liberated, hence the distinct rise of temperature which occurs.

Physical effects of increase or reduction in amount of heat present.—The liquefaction of a solid and the freezing of the same substance from a liquid form, take place at a constant temperature with small exceptions

due to abnormal conditions of pressure, purity, and stillness. In the same way the vaporization of a liquid and the condensation of the same substance into a liquid form, take place at a definite temperature for each pressure, but varying largely for different pressures.

Specific heat.—The amount of heat which is required to raise the temperature of one pound of any substance through one degree is called the specific heat of the substance at the temperature and in the condition in question. In ordinary English practice it is always given in British thermal units, expressed or implied. The specific heat of a substance varies with its condition; that of ice is about half that of liquid water, and that of steam is rather less still.

Table I. gives the following data for some of the most important substances used in engineering work—

(*a*) Temperature of melting or freezing.
(*b*) „ „ boiling or of liquefaction from vapour.
(*c*) Latent heat of liquefaction.
(*d*) „ „ vaporization.
(*e*) Specific heat in solid form.
(*f*) „ „ liquid form.
(*g*) „ „ form of vapour.

Variations in data.—In some cases the figures vary largely under different temperatures and conditions, consequently, for exact figures, special works should be consulted. In addition to the well-known treatises on heat and natural physics, Professor Everett's *Units and Physical Constants*, and Miller's *Chemical Physics* may be mentioned. The specific heats of gases are stated in the usual form applicable to gases under constant pressure, and therefore able to expand freely on a rise in temperature. Under such conditions gases which

TABLE I.—HEAT DATA OR CONSTANTS.

	Temperature.		Latent heat.		Specific heat.		
	Melting.	Boiling.	Liquefaction.	Vaporization.	Solid.	Liquid.	Gaseous.
	a. F°	b. F°	c. British Thermal Units.	d.	e. B.T.U.	f. B.T.U.	g. B.T.U.
Wrought Iron					·1124		
Cast Iron ...	2200	Under atmospheric pressure					
Copper	1930				·0933		
Tin	450		25·6		·0559	·064	
Zinc	680		50·6		·0935		
Lead	620		9·7		·0315		
Mercury	— 40	662	5·		·0319	0·335	
Gold	1913				·0316		
Silver	1750		38		·0559		
Platinum ...	3227		49		·0323		
Aluminium ...					·2185		
Glass (variable)	2000				·198		
Water	32	212	143	965	·504	1·000	·4805
Wood Spirit ...		150		475		·600	·4580
Alcohol		172		364		·585	·4534
Ether		95		163		·529	·4796
Bisulphide of Carbon ...		115		156		·238	
Turpentine ...	14	319		133	·416	·426	·506
Linseed Oil ...		700					
Perchloride of Tin		240		55			·094
Air							·2375
Oxygen		– 292					·2175
Nitrogen ...							·2438
Hydrogen ...							3·4090
Carbonic Acid							·2163
Sulphuretted Hydrogen ...							·2432
Sulphurous Acid							·1540
Ammonia ...							·5084

are exposed to atmospheric pressure perform work in displacement of the atmosphere. Conversely, when such gas is cooled the atmosphere closes upon it, and in the act performs the same amount of work in restoration of the original condition. The work expended in displacement of the atmosphere causes the disappearance of a quantity of heat, so that a greater

total quantity is required to raise the temperature one degree, than would be the case if the gas were prevented from expanding by heat. In other words, the specific heat for constant pressure is greater than that for constant volume.

This point may be illustrated by means of a cubic foot of air, which weighs ·080 pound at a temperature of 60° F., or 520° F. above absolute zero, and at mean atmospheric pressure, as defined by means of a mercury column 30 in. in height. If the temperature of this air be increased by 520° or to 580° F., its volume will be doubled. The specific heat of air under constant pressure is ·2375; therefore $\cdot080 \times \cdot2375 \times 520 = 9{\cdot}880$ units of heat required in the operation. But mechanical work has been performed in the process equal to 2117 foot pounds. Dividing this by the mechanical equivalent of heat (772), it is found that the mechanical work done accounts for $\frac{2117}{772} = 2{\cdot}742$ units. It follows that $9{\cdot}880 - 2{\cdot}742 = 7{\cdot}138$ units of heat have been expended in causing the actual increase of temperature, which is only about 1·5 per cent. greater than the results obtained by the use of the specific heat of air under constant volume, which has been independently found to be ·169. If two cubic feet of air at the higher temperature were to be compressed so as to occupy only one foot, the air will rise in temperature by virtue of the heat expended in producing the mechanical work. If allowed to expand and perform mechanical work, it will resume its former temperature. If between the compression and the expansion it is subjected to cooling, to the temperature at which it existed before compression, the temperature will still fall during expansion. The study of the various questions dependent upon specific heat or related thereto is worthy of much attention. These include many points in the

analysis of heat in engine trials, the production and use of compressed air, and the combustion of fuel under high pressure, with the object of attaining specially high temperatures by direct means.

Various conditions of water due to changes in heat.— The successive physical effects of heat continuously applied to water as a typical substance furnish an example of considerable interest. Heat may be assumed to be supplied at the uniform rate of one unit per second, and the whole subjected to mean atmospheric pressure. One pound of ice at 0° heated at this rate will rise in temperature until it reaches 32°. In this process the amount of heat required is $32 \times \cdot 504 = 16\cdot 128$ units, and the length of time will be the same number of seconds. The whole will remain at 32° until completely melted. For this purpose 143 units of heat, being equivalent to the latent heat of liquefaction, will be required, and a corresponding length of time occupied. The volume of water produced will be about one-tenth less than that of the ice producing it. While the temperature increases from 32° to 39·1°, the water contracts slightly, and at the latter temperature assumes its condition of maximum density. The further application of heat causes a nearly constant increase in temperature and an increasing rate of expansion until 212° is reached. The quantity of heat required to raise the temperature one degree increases slightly, so that 180·9 units are required to cause a rise from 32° to 212°. When the temperature reaches 212° the water evaporates with some commotion, forming steam, the temperature remaining constant during the operation. The change takes place by reason of the absorption of the latent heat of evaporation, amounting to 965 units per pound of water at mean atmospheric pressure. This process will occupy 16 minutes 5 seconds.

If the mass is subjected to additional pressure the temperature will rise to a higher point before the boiling point is reached, and the latent heat also varies. Steam which is in contact with water cannot be caused to assume a temperature above or below a definite one, corresponding to the pressure as set forth in table, *b*. Steam which is completely vaporized may be raised in temperature by the continued application of heat, under either constant pressure or volume, but either one or both must increase.

Evaporation of water exposed to atmosphere.—At all temperatures, liquid water and ice in contact with dry air suffer evaporation, the rapidity of which increases very much as the temperature rises. Air which already contains moisture absorbs additional moisture with diminishing activity until each cubic foot contains the weight of a cubic foot of steam as given in Table IX., when it is said to be saturated. Heat is absorbed in proportion to the amount evaporated.

Partial condensation of steam.—In all cases in which the addition of heat produces a change of temperature or condition, the reversal of such changes will be accompanied by a liberation of heat exactly equal in amount. When saturated steam is subjected to loss of heat by radiation, or by reason of mechanical work performed by it, a precisely equivalent amount of condensation occurs, the water produced becoming separated from the steam or remaining diffused in it according to its quantity. Steam which contains liquid water diffused or suspended in it is called "wet steam," and possesses properties different from those of dry saturated steam. When a liquid is caused to evaporate, by a sudden or gradual reduction in the pressure to which it is subject, the heat to furnish the latent heat of evaporation must be furnished by the liquid, or by the walls

of the vessel containing it, or by neighbouring bodies, one or more of which must suffer a corresponding reduction in temperature.

Transfer of heat.—When heat is imparted by one body to another, the quantity possessed by the first is reduced equally with that acquired by the second. It may be transferred in either one or both of two ways. It may suffer conduction by actual contact, directly or through the interposition of another body, or it may be transferred by radiation, for which purpose no sensible intervening medium is necessary, but only the ether which pervades all space. Radiant heat is transmitted at a velocity of 186,000 miles per second. The transmission of heat in furnaces and flues takes place in both ways, but the solid fuel and the minute solid particles in flame impart most heat by radiation, and the non-luminous gases in the flues by conduction. Radiant heat may or may not be accompanied by light rays. It may pass uninterruptedly through a solid, liquid, or gaseous substance, exactly as light passes through glass. In this respect the term "diathermancy" corresponds to transparency for light. Radiant heat may be reflected in the same way as light. The radiation of heat from a solid body is largely affected by the condition of the surface. A smooth polished surface will radiate much less heat than a rough one, and as a rule a black surface is a better radiator than a white one. Roughness as affecting this question is not such as admits of absolute measurement, but rather a molecular roughness, such as that of a lampblack-covered surface. A surface which radiates heat freely also absorbs heat with equal facility.

All bodies conduct heat.—All bodies possess the power of conduction of heat. But those in which the power is comparatively low are said to offer resistance to the

passage of heat, or to be non-conductors, which expression must be understood in a relative sense, as an absolute absence of conducting power cannot be attained. The amount of heat transmitted through a body varies with its nature; also in direct proportion to the difference of temperature on the two sides, and usually in inverse proportion to its thickness. The transmission of heat through metal plates of moderate thickness such as used for boiler plates, appears, however, to be practically independent of thickness and of the metal employed, but to vary largely with the condition of the surfaces.

Expansion of bodies by heat.—With a few notable exceptions—that of water has been already referred to—all bodies expand by reason of increase in temperature. A bar of iron or steel 900 inches in length will expand one inch if its temperature is raised from the freezing point to the boiling point of water (32° to 212° F.), the amount of expansion being proportionate to the length and to the rise in temperature. Different substances vary exceedingly in their ratio of expansion, and that of a liquid is much greater than that of the same substance in solid form, while, with few exceptions, that of the same in a gaseous form is greater still. That of a solid is practically uniform at all temperatures, but varies with its condition of crystallization or otherwise. Some bodies of parallel, fibrous, or analogous structure expand differently in different directions. The ratios or co-efficients of expansion of solid bodies are usually referred to their linear dimensions. In some cases they are referred to cubical capacity, in which case within moderate limits the co-efficient of a homogeneous substance is practically three times as great as the co-efficient of linear expansion. The co-efficients of expansion of liquid or gaseous bodies by heat are always

given with reference to cubical capacity or volume. Those of liquids increase as the temperature rises. The expansion of a newly-formed vapour is somewhat erratic, owing to the fact that it usually contains small particles of the liquid incompletely vaporized. But all substances in perfectly vaporized condition expand equally, and in exact ratio to their absolute temperature, which is obtained by the addition of 460 degrees to the reading on Fahrenheit's scale. Thus one cubic foot of a perfect gas at 40° F. occupies one and a fifth feet at 140° ($1 \times \frac{140+460}{40+460} = 1{\cdot}2$ cub. ft.). This is true when the gas is allowed to expand freely under constant pressure. If the volume is maintained constant, the absolute pressure will rise in the same proportion.

Convection.—The weight of all bodies remaining constant, their expansion by the addition of heat, obviously causes a reduction in density or specific gravity. In solid bodies this is, for ordinary purposes, of no importance. But when a mass of liquid or gaseous matter suffers local heating or cooling, the equilibrium is disturbed; hot light matter ascends, or tends to ascend, and cold matter to descend. This phenomenon is called "Convection," and being of some importance is treated in a separate chapter.

Conversion of heat into work.—Heat may be changed into mechanical work, and conversely mechanical work may be transformed into heat. Facts have already been stated which bear upon the first part of the proposition, but the latter part is more easily demonstrable. The school-boy produces startling effects by means of a brass button vigorously rubbed a few seconds. A machine of which the lubrication is neglected becomes hot. When air is compressed its temperature rises considerably. The temperature of a quantity of water contained in a vessel may be raised by mechanical

agitation. The first two instances suggest means whereby sufficient heat might be generated for application to the boiler of a steam-engine, not with financial or practical success, but as a physical possibility. In the third case, the heat is developed in connection with a substance of a high degree of elasticity, and the mechanical work expended can be at once recovered by a simple reversal of the process. The fourth example provided the means whereby Dr. Joule permanently established and quantitatively expressed the relation between heat and mechanical work. Most of the mechanical work expended throughout the world is ultimately consumed in overcoming frictional resistance, and therefore re-appears as heat, though so much dispersed as to escape all ordinary and direct means of estimation. Some is expended in raising weights, which is a means of storage of work to be restored at any time; in many cases this work will ultimately be directly expended upon frictional resistances. Pumping engines perform work of this class which in practically every case is directly or indirectly accounted for by frictional resistance.

Extinction of heat equivalent to work produced.—In all heat engines, such as steam engines, gas engines, hot air engines, etc., heat disappears, and an exactly equivalent amount of mechanical work or power is produced. In every case heat is rejected by the engine and lost. If a perfect engine could be made, the whole of the heat would be converted into work, and no accumulation of heat in the atmosphere, the water used, or any surrounding objects or substances would occur, whether the engine were or were not in use. In practice this is never secured, and probably never will be, but every improvement tends towards the attainment of such a condition. Mechanical refrigerating machinery

is adopted for the purpose of withdrawing heat from contiguous substances. The heat thus withdrawn is not utilized by the engine for the production of mechanical work, but is rejected, in addition to the amount of rejected heat due to the production of the power necessary for the operation of the engine. Air or other gaseous fluid is compressed by mechanical means, and thereby heated. It is then cooled while still under pressure, and expanded in such a manner as to produce work, in which process its temperature suffers reduction in proportion to the amount of work done. The cold gas is then applied for the abstraction of heat from other matter.

Mechanical equivalent of heat.—Rumford and Davy, at the beginning of this century, laid the foundation of the principle connecting mechanical work and heat. Others took part in the study of the question, but Dr. Joule of Manchester, about 1847, experimentally proved the practical quantitative relation of the two forces. He applied a paddle to set in motion a known quantity of water contained in a vessel. The motion of the paddle was produced by means of a weight actuated by the force of gravity. The work due to the falling weight was accurately measured, as also the increase of temperature in the water caused by the work done. He showed that one British thermal unit of heat was produced by means of the expenditure of an amount of mechanical work measured by 772 foot-pounds. This he corroborated in other ways, and it has been shown, by the expansion and compression of gases, that the converse action occurs wherein the expenditure of one unit of heat is accompanied by the production of the same amount of work. The term "Joule's Mechanical Equivalent of Heat" is applied to a quantity of work measured by 772 foot-pounds. In 1878 Dr. Joule

repeated the investigation of this question, and proved that, according to his thermometer, the increase of one degree was obtained by the expenditure of 772·55 foot-pounds. He subsequently discovered the thermometer to be inaccurate to such an extent that the corrected mechanical equivalent became 775·47 foot-pounds. The difference is, however, so small that for practical purposes the original figure of 772 is always adopted.

Water usually adopted as the medium in conversion of heat into work.—In the conversion of heat into mechanical work, the heat is applied to effect an increase in the volume of some substance. Water is usually employed, and is changed from a liquid to a gaseous condition, or steam. In this process a very large proportion of the heat expended is due to the high latent heat of vaporization of water, which, in almost all cases, accounts for over seventy per cent. of the total amount of heat imparted to the fluid. In a direct sense this appears to be a loss, and many proposals have been made for avoiding such loss by the adoption of fluids whose latent heats of vaporization are lower than that of water. But it is found that condensation of steam takes place in the cylinders of the steam engine to a very undesirable extent, which condensation would be increased by the adoption of a fluid of lower latent heat. Consequently, apart from any question of abundance or cost involved in such a change, it would appear that little or no gain can be hoped for in such a course. This does not apply to the adoption of combined engines in which ether or other fluid which boil at a low temperature are evaporated by means of the exhaust steam from a steam engine. In this way, condensation of the steam is promoted and additional power produced, but whether the amount of this power

is sufficient to give a commercial return for the additional outlay and maintenance of machinery, and for the replacement of waste in the supplementary fluid, remains to be proved.

Steam of high pressure efficiently expanded gives better results than steam of low pressure, one reason for which consists in the fact, that a less amount of water is necessary for the performance of a certain amount of work. Thus the proportionate and absolute amount of heat expended as latent heat is very much reduced.

Air occasionally adopted as medium.—By the use of air as the motive fluid in "Caloric Engines," all expenditure upon the item of latent heat is avoided, and herein lies a great advantage possessed by them. Cylinder condensation is also entirely absent. But the volume of air pumped in is much greater than that of water pumped into a steam boiler, and therefore more power is absorbed in the operation. Much difficulty has also been experienced in connection with the wear and tear of the machinery and appliances at the high temperatures to which they are exposed. But in the modern gas engine, the superior facility for control presented by gaseous fuel, a fortunate combination of cylinder for development of motive power, with pump for admission of air, and the introduction of lubricating oils adapted to endure high temperatures, have caused the principle of the caloric engine to assume a form eminently adapted for use under many conditions of practical work. An engine has recently been experimented with at Krupp's works, in which powdered fuel is supplied to a cylinder with a suitable proportion of air, and exploded as in a gas engine.

Chemical separation.—Heat may be expended in the chemical separation of elements which possess a strong

affinity for each other; and, conversely, it may be reproduced when the same bodies are caused to recombine. The heat which is absorbed in the separation of carbon and oxygen, either directly by growing plants, or indirectly by chemical reactions or electrical means, is precisely equal to that produced in the chemical combination of the same substances, or combustion.

Electrical separation.—Mechanical work may be transformed into electrical force or energy by means of a dynamo-electrical machine. The same machine may also be used to re-transform electrical power into mechanical work. Electrical power may also be generated in a galvanic battery by the consumption of zinc or other material.

Mutual relation of different forms of energy.—The absolute laws, to which the several forces are subject in their relations with each other, may for the purposes of the present work be sufficiently illustrated by considering the changes which take place in a quantity of zinc. When this suffers oxidation a definite amount of heat is generated, in a manner similar to the generation of heat when carbon is oxidized or burnt. If the oxide is reduced to the condition of metallic zinc by the agency of heat, the same amount of heat will disappear as was developed in the oxidation. The metal may then be consumed in a battery generating electricity, which may be transformed into mechanical work by means of a dynamo-electrical machine. The mechanical work may be applied in overcoming frictional resistance, in such a manner as to produce heat, the exact amount of which will be less than the quantity originally absorbed in the reduction of the zinc from its oxide, by an amount precisely equal to the sum of the small losses which arise at each stage of the circuit. This type of action, or series

of actions, may be illustrated by an infinite number of examples, each of which would invariably show that it is impossible to make an expenditure of energy in one form, and in return receive more than a precise equivalent in any other form. Still less is it possible to command a supply of energy without any expenditure at all.

CHAPTER II.

FUEL AND COMBUSTION.

Heat practically derived from combustion of fuel.—All heat utilized in connection with industrial operations is derived from the combustion of fuel in a furnace. This is a chemical process of combination between the combustible constituents of the fuel and oxygen, which forms about 23 parts by weight, or 20·81 parts by volume in every 100 parts of the atmosphere over the whole of the earth's surface.

Combustion of carbon in fuel.—In almost every kind of fuel carbon is the element of greatest importance. Its powerful chemical affinity for oxygen causes combination when the two are brought into contact at a high temperature, with the production of great heat in the process, which continues until the supply of one element becomes exhausted. In this process carbonic acid gas, or carbon dioxide, is produced, which always contains 12 parts of carbon and 32 parts of oxygen by weight. If, however, the supply of oxygen should be limited, the carbonic acid gas immediately takes up 12 additional parts of carbon, forming two volumes of carbonic oxide gas, or carbon monoxide. This additional carbon can only be absorbed when it exists in an incandescent state. If while the carbonic oxide remains

at a high temperature, it comes into contact with oxygen, a further quantity of the latter will be absorbed and carbonic acid again produced. These alternative actions may be repeated indefinitely, but in no case is any compound produced from carbon and oxygen, intermediate between the two described, or different from these. But, according to the relative proportions of the two constituent elements, a mixture of carbonic acid and carbonic oxide may be produced.

Amount of heat produced in combustion of carbon.—The amount of heat produced in the combustion of one pound of carbon has been accurately determined in a large number of experimental investigations to be 14,400 British thermal units, and more recently, by the use of improved apparatus, to be 14,652 units. This never varies for pure and dry carbon. The heating power of fuel is often expressed by the number of pounds of water which can be evaporated by one pound of the fuel. To avoid confusion this is assumed to take place at a temperature of 212° F. and mean atmospheric pressure, and that no heat is absorbed in raising the temperature of the water or of the steam. Under such conditions one pound of water is raised into steam by the addition of heat amounting to 965 British thermal units. The quantity of water evaporated in this way by one pound of pure and dry carbon is 15·18 pounds. If, however, while still in contact with the burning fuel, the carbonic acid gas is allowed to become reduced to the condition of carbonic oxide, the additional quantity of carbon absorbed is lost. In this process sensible heat disappears, by reason of the additional amount of carbon gasefied; hence the consumption of two pounds of carbon in this way yields a less quantity of heat than does the complete combustion of one pound. One pound of carbon burnt to produce

carbonic oxide will furnish heat for the evaporation of 6 pounds of water. By the supply of additional air, under suitable conditions, the carbonic oxide becomes changed to carbonic acid, and heat is generated, which, together with that produced in the first instance, amounts to a total precisely the same as though the same quantity of carbon had been completely burnt to form carbonic acid in the first instance.

Loss of heat by imperfect combustion.—In puddling and other metallurgical furnaces, much heat is often lost by reason of the formation and escape of carbonic oxide. In many such cases this takes fire at the top of the chimney, producing great heat, but the flame is so faintly luminous that it may escape notice by daylight. The same action often occurs in crane boilers, donkey boilers, and small marine boilers, by reason of excessively thick fires, badly-proportioned fire-bars, dirty grates, or other causes which obstruct the supply of sufficient air to support perfect combustion.

Production of carbonic oxide gas.—Carbonic oxide gas is the most important constituent of the gases generated from coal and other fuels in "gas producers," and hence called "producer gas." This is largely used in various metallurgical processes, in gas engines, and occasionally for the firing of steam boilers. The waste gases from blast furnaces also contain very large quantities of this substance, which is burnt to useful purpose in connection with steam generation and heating of the blast. Large quantities of this gas are very rarely found in the gases in the chimneys of ordinary boilers. When this does occur, there is a corresponding direct loss in economy; and owing to the additional weight of gases to be dealt with, the chimney draft is somewhat impaired.

Combustion of hydrogen.—Hydrogen is the element

in fuel which is next in importance to carbon. In all cases 1 part by weight of hydrogen unites with 8 parts of oxygen to produce water-vapour or steam. In the combustion of one pound of hydrogen, the heat generated amounts to 64,800 British thermal units, and is sufficient to evaporate 67·15 pounds of water from and at 212° F. This amount of heating power is very much greater than that of carbon, but the large amount of water produced, and of nitrogen remaining from the air after the removal of its oxygen, prevent the furnace temperature from rising in proportion to the absolute heating power.

Combustion of hydrocarbons.—The hydrogen contained in most fuels exists in combination with carbon, forming hydrocarbons, or carburetted hydrogens of variable composition. These are easily evolved on the application of heat. Some, when separated, remain as perfect gases at all ordinary temperatures, while others are condensible as tar or pitch. Of the former the most important is "marsh gas," which contains 4 parts of hydrogen by weight and 12 parts of carbon. This is largely contained in illuminating gas, and in the natural gas which in some parts of the world is drawn from the earth, and forms a most convenient fuel. The "fire damp" of the miner is largely composed of marsh gas. "Olefiant gas" is the hydrocarbon next in importance to marsh gas. It contains 4 parts of hydrogen united with 24 parts of carbon. This also is contained in coal gas, and it burns with a more luminous flame than that of marsh gas. The more condensible hydrocarbons are more variable and complex in constitution, but very much resemble the above as regards their combustible properties.

Instability of hydrocarbons.—All the important hydrocarbons are capable of distillation at a tempera-

ture below that at which they will suffer complete combustion, especially if the air is limited in amount or imperfectly mixed. But under such conditions they undergo decomposition; the hydrogen, being the more actively combustible element, is burnt, while some of the carbon escapes in the form of smoke, the black appearance of which is imparted by carbon in a state of the finest division. Under such conditions also olefiant gas surrenders half of its carbon to form black smoke and marsh gas. When the gases are uniformly mixed with the required quantity of air to effect combustion, and the temperature of the whole maintained at or above 2000° F., the combustion is perfect, and no smoke is produced.

Intimate contact necessary for securing perfect combustion.—The chemical combination of the atmospheric oxygen with the combustible matter in the fuel, can obviously only be accomplished when the two are in contact. The influence of close contact is shown by the difference in the intensity of combustion displayed in the use of air and of pure oxygen. In the first case, the combustion is impeded by the presence of nitrogen, which forms the greatest part of the atmosphere, and which is absent in the second case.

Use of a fire-grate.—Fuel is generally burnt upon a grate, composed of separate bars set on edge, with spaces between for the passage of the air requisite for the support of combustion. In the design of a grate, the most serious practical difficulty encountered consists in the provision of sufficient space for the passage of air, without sacrificing the strength of the bars or allowing unburnt fuel to drop through.

Supply of air to fire.—About 12 pounds of air are necessary for the actual combustion of each pound of coal of good average quantity. It is, however, practi-

cally found to be impossible to supply the air so as to ensure complete combustion, unless the air is in excess. For this reason about 18 pounds of air must be supplied per pound of coal with hand firing, even when performed with the greatest care, while 24 pounds is quite commonly supplied. If the whole of the latter amount should be admitted through the grate, and the area of openings is one-fifth that of the whole grate, including cross-bars, while 20 pounds of coal are burnt per hour, per square foot of grate surface, the velocity of air passing through the grate would not exceed 8 feet per second. But the openings through the grate are always very much obstructed by fuel, and they are often allowed to become additionally so by the accumulation of clinker and ash. On account of such obstruction it is only under the most favourable circumstances, and by the exercise of great care, that the supply of air is efficiently maintained. Many attempts have been made to improve the air supply through the grate by means of an increase of area, obtained by means of sinuosities intended to secure an increase in length of opening while maintaining the usual width. Many such arrangements are based upon erroneous views as to mensuration. Others would answer very well with coal free from clinker and ash, but such coal is practically unavailable; otherwise, much of the difficulty experienced with the usual pattern of grate bar would disappear. In practice they are found to be exceedingly difficult to keep clear, and therefore cannot be considered to be successful. In Martin's grate the bars are made continuous, and supported by narrow cross bearers, so as to give good clear space from end to end, free from the usual interruptions. They are arranged to be slightly movable, separately, so as to break down the clinker. This grate is said to have proved very successful under difficult

conditions, especially when in combination with Martin's furnace doors. Perret's furnace—made by Messrs. Bryan Donkin and Co.—is arranged for use with forced blast. The bars are only about five-eighths of an inch thick, so as to give a large number of interspaces, which are only about one-tenth of an inch in width, so that the area for admission of air is only about one-seventh part of the total grate area. The bars are very deep, and are kept cool by the application of a bath of water applied to their lower edges, by which means the ash produced is at once cooled on touching the bars, and is prevented from sealing the openings. The bars are consequently kept clean, open, and in good efficient condition, though covered with ash. This grate is especially useful for the consumption of cheap fuel which contains much ash. The use of forced blast to a pressure equal to the chimney draft causes an increase in the amount of air supplied through the grate, in the ratio of $\sqrt{2} = 1{\cdot}41$, and promotes perfect combustion in the mass of fuel. In this way heat is not radiated so freely from the fuel, and the temperature rises, which further improves the combustion, in the same way as does a fire-brick arch over the fire, which is provided in special cases.

The use of a jet of steam below the fire also largely prevents the sealing of the bars by hot clinker. When steam comes into contact with such matter it suffers decomposition, heat disappears, and the hot matter becomes cold and incapable of closing the grate. The heat so absorbed is not lost, but re-appears when the oxygen and hydrogen subsequently re-combine and form water, subject, however, to deduction on account of heat carried up the chimney by the produced water. It is impossible to give too much freedom for the access of air through the grate bars. But it is useless to

provide area for this purpose under conditions which practically prevent its preservation in a clear state. The ordinary flue damper provides an efficient check against the admission of too much air in this way, and an ash-pit damper may be provided for a similar purpose. But when the air supply is cramped, the furnace is always working under difficulties, which often fail to be appreciated.

Air is sometimes caused to pass in the reverse direction—*i. e.* downwards through the burning fuel—with a view to ensure perfect combustion. The object is, however, imperfectly secured, and the passage of the heated gases through the grate leads to very great wear and tear.

Conditions of stoking.—If a solid piece of fuel is placed upon a grate in an incandescent condition, surrounded by air, it is clear that only the film of air in contact can enter into combination with the substance of the fuel. If a number of such pieces are spread apart upon the grate the same will occur in each case. These may be supposed to be so adjusted that the spaces will exactly suffice to pass air for perfect combustion. But this condition is soon disturbed by the burning of the fuel, so that the pieces are reduced in size and the spaces enlarged. The same also takes place when the fuel is applied in layers. At one place the resistance to the passage of air is reduced, an excess of air is admitted, a hole is formed, the gases are decomposed by cooling, and fuel is wasted. At another place where the fuel is less freely combustible, the air passage is obstructed, combustion languishes, and a complete stoppage is caused. The difficulty which is practically experienced in correctly adjusting the air supply to the chemical requirements of the fuel is therefore an important one. Perfect combustion would

be possible with a thin fire if it could be kept quite uniform. But in a thick fire the fuel to a great extent automatically adjusts itself, so as to promote uniform combustion, and in this way a thick fire is usually more efficient than a thin fire. Thick fires, however, produce large quantities of carbonic oxide by the reduction of carbonic acid as it rises through the hot fuel. The admission of an accurately proportioned supply of air above the fire will complete the process by the production of carbonic acid, provided that a sufficiently high temperature is maintained. But when carbonic oxide is produced in large quantities, heat is absorbed to such an extent that the upper part of the fire is cooled and combustion checked, causing loss of carbonic oxide. The temperature may also be sufficiently great to cause the distillation of the hydrocarbons contained in the fuel, but not sufficiently so to cause them to be completely burnt. These, therefore, pass away unchanged, or more frequently are decomposed, so that black smoke is formed, carried forward by the furnace gases, and discharged into the atmosphere. If, however, the temperature of the fire is kept sufficiently high to maintain a bright flame above it, and an accurately adjusted supply of air is admitted above the fire, the production of smoke may be avoided. But the supply of air above the fire may be most accurately adjusted as to quantity, and still fail to effect its intended purpose. Such air may be allowed to pass along the furnace in a continuous and sharply-defined mass above the gases from the fire. The latter being at a higher temperature tend to rise above the heavier air, but by a strong chimney draft are prevented from doing so before passing the bridge. The air and the combustible gases must be brought into contact, particle to particle, before they will enter into combination. In many locomotive

boilers, and in some stationary boilers, this is largely effected by means of a deflecting plate which runs downwards from the upper part of the firing doorway, to project over the fire. Such deflecting plates are usually fixed, in which case they must allow sufficient space for firing. Martin's furnace doors form movable deflecting plates, which may be closed to exclude the air or adjusted as required. The curve is carefully designed to direct the air into close contact with the fuel. This causes the air and gases to become thoroughly intermixed, and the production of smoke to be very largely prevented. When these doors are used, a large proportion of the fuel is burnt on the upper side of the fire, and when very small coal is used, the draft is liable to carry ashes into the flues and up the chimney. These doors have been very largely used, and have proved exceedingly successful. When a fixed deflecting plate is used, its distance from the fire must allow space for stoking, and thus is too great to secure such a complete action as that of Martin's doors, and there is no object in adopting great refinement of curve, as some eddying action is useful in promoting the mixture of gases and air. Such admixture is in many cases secured by means of the supply of air in small jets or threads. Air is effective in this way in proportion to the fineness of division which is secured. But by attempting to carry it too far the jets become weakened. This point is again referred to in connection with forced draft. Air is often supplied at the bridge of a furnace. But the gases have been partially cooled before arriving at this point, and perfect combustion is more difficult to secure than by judicious admission of air at the furnace door.

In many cases smoke is produced by reason of carelessness or indifference on the part of the stoker, in

adding excessive quantities of coal at one time. No means exist whereby smoke may be prevented when a fire is smothered in this way, as the temperature is reduced and the draft quenched. The steam production of a boiler thus treated is very largely suspended until time is allowed for the fire to burn brightly. Such a boiler, if accurately proportioned to its work, must suffer from forcing at other times in order to maintain steam. Solid fuel of any kind containing hydrocarbons is liable to produce smoke if placed on the furnace in large pieces. Under such conditions, the pieces crack, and give off gases which are quite unacted upon in ordinary furnaces, though in the furnaces of locomotives running at full speed the draft, the temperature, and the motion of the fuel may suffice to cause efficient combustion. In the furnaces of stationary boilers working under natural draft, large blocks of coal deflect the air out of contact with much of the surface. About 1785, Smeaton, in giving instructions as to the operation of an engine at the York Water-Works, directed that every coal larger than a goose's egg should be broken, and that fuel should be added to the fire frequently and in small quantities. These instructions are very good, but would be improved by a reduction in the standard of size allowed. With equal care, small coal is found to produce less smoke than large coal from the same seam. Large coal fetches a higher price, but possesses no special value or advantage for ordinary work, except that it is more convenient in kindling fires, it is better adapted for storage owing to the small surface exposed in comparison with its weight, and that it usually contains less dirt than does small unwashed coal.

In the most simple manner of stoking, the coal is spread over the grate with as much uniformity as is deemed to be necessary, and allowed to remain

untouched, fresh coal being added from time to time. This answers well for anthracite and coke, also for all coal of small sizes, provided that the fresh coal is added in small quantities. The application of a slice or rake is useful for levelling the fuel and preventing the formation of holes, but should be practised to the least possible extent on account of the great liability to produce smoke. Bituminous coal, and other fuels liable to produce smoke, are sometimes spread more freely upon the part of the grate nearest to the door. The rake is used for levelling, and also for pushing the partly-consumed fuel towards the back of the fire. Any smoke which may be formed at the front of the fire is carried over the hot fuel behind. If this is done in such a manner that the two are brought into contact, the smoke which has been already produced may be really burnt. This is, however, a matter of some difficulty, to effect which the air should be admitted in such a manner as to pass along the whole length of the fire, in contact with it, sweeping the gases along. A hanging water bridge or brick arch is sometimes adopted to assist the operation. These give good results while in good order, but are costly to maintain. The hanging bridge provides an unimportant increase in the amount of heating surface of the boiler. "Smoke burning" is a very delusive expression, frequently used but seldom realized. The prevention of smoke production is much more easily achieved, but even this is only to be secured by great attention, unless a large excess of air is admitted to the furnaces. The thickness of the fire should be kept as uniform as possible, or slightly thicker at the edges. When allowed to become higher in the centre the risk of formation of holes at the sides is increased, when, in addition to other evils, the plates of the boiler are very apt to be chilled by cold air.

Production of smoke.—The carbon, which gives its black appearance to smoke, is obviously lost as regards its combustible value. The amount of this is, however, much less than would be naturally expected, which is accounted for by its extreme fineness of division. M. Scheurer-Kestner found that the total loss caused by the escape of carbon in very dense black smoke amounted to only 1 per cent. of that in the fuel consumed. The production of smoke is often accompanied by a thick, sooty deposit upon all heating surfaces, which interferes with their efficiency, though a thin one promotes absorption of heat. Black smoke is usually accompanied by the production and loss of carbonic oxide, and sometimes of hydrocarbons. Each is produced by similar causes and prevented by similar measures.

The production of smoke from a slowly-burning fire may be prevented by an increase in the amount of air admitted by the fire door, but the amount necessary to effect this purpose, in the absence of a deflecting plate or its equivalent to direct the air against the fire, is very great.

The free carbon contained in smoke is a good radiator of heat, and by its presence facilitates the transfer of heat from the gases to the heating surfaces, but the degree in which this affects the general balance of economy is not exactly determined.

A good sharp draft causes much more perfect combustion than a sluggish one, by promoting more intimate contact between the air and the solid or gaseous fuel, just as the water in a rapid river becomes more quickly and completely mixed than does that in a sluggish river. A boiler which burns fuel at a low rate per square foot of grate surface, but for which a good chimney draft is available, will work more efficiently if part of the grate is blocked, so that the energy of combustion is

increased. A lamp without a chimney burns with a smoky flame, but on the application of a chimney the character of the flame suffers a complete change, becoming much smaller, brighter, and more energetic than before. It is also quite free from smoke, and clearly shows the importance of a good draft. Forced draft, or the assistance of the draft by mechanical means, is often resorted to, with the object of improving the combustion, or increasing the amount of heat obtained in the furnace, by reason of a proportionate increase in the amount of fuel consumed, as well as for the reason already given in connection with Perret's grate.

Injury caused by admission of air during stoking.—During all stoking operations, care should be taken to allow the entry of the smallest possible amount of air through the furnace. Air thus admitted exercises a cooling effect upon everything with which it comes in contact, to the structural detriment of the boiler, bridges, and brickwork; it also carries away much heat to no practical purpose.

Use of mechanical stokers.—The many elements of uncertainty and irregularity in connection with hand-firing with solid fuel, render impossible the accurate adjustment of the air supply to the amount necessary for perfect combustion. The powerful cooling action of the fresh fuel and of the open fire door, which are more or less inseparable from hand-firing, are avoided by the adoption of mechanical stokers. These give a uniform supply of fuel in one of two ways, to some extent ruled by the character of coal used. In the first way, the fuel is fed from a hopper in such a manner that it first undergoes a process of coking at the front end of the grate, after which it is propelled forward by means of alternate movements of alternate bars of the grate. In the second class of mechanical stokers, the coal is sup-

plied from a hopper of arrangement very similar to the first. But the distribution over the grate is effected with more or less uniformity by means of a paddle, piston, or other detail for throwing and spreading, or "sprinkling," the coal. This purpose is obviously very difficult of attainment with the requisite degree of accuracy. Some degree of economy in labour is effected by the adoption of mechanical stokers, but the amount of this is very much a matter of opinion. They are, however, generally adopted with a view to the abatement of smoke production, in which as a rule they are very successful. They are obviously most suitable for application in cases where the work of the boilers and the coal used are most uniform. In coking stokers, the ashes are carried forward and deposited at the back end ready for raking out. In some cases air is very liable to be admitted at this point in large, irregular, and uncontrolled quantities. When sprinkling stokers are adopted, the cleaning of the fires is rather difficult. So long as everything is in perfect order, and irregular leakages of air prevented, the supply of air to the burning fuel may be adjusted with very fair accuracy. But in all cases there is considerable difficulty experienced in ensuring that the grate shall remain entirely covered with fuel, and the formation of holes avoided. For attention to this matter, convenient doors should be provided, so that every part of the grate can be reached by a slice or rake; and it is no less important that the same should be frequently and judiciously used. Leakages of air also often arise by reason of defective fittings at the furnace fronts and about the grates. Hence an attendant, whose skill and faith in the performance of his work are above question, is fully as valuable as for hand-firing. A man of opposite character may not embroil his employers by

reason of offences against the smoke laws, but he will allow large amounts of heat to escape up the chimney in air which is unnecessarily admitted. When mechanical stokers are adopted there is even a more urgent necessity than is otherwise the case for the adoption of constant sampling and analysis of furnace gases. The action of mechanical stokers when in *perfect* order, may be profitably studied as embodying the principles to be followed in the practice of hand-firing, with the exception that they are often liable to the production of a "haycock fire," which in all kinds of furnaces is very objectionable.

Loss of heat caused by supply of air in excess.—If it were possible to usefully extract the whole of the heat from the furnace gases, and discharge them at the temperature at which the air was originally received, no direct loss would be incurred by reason of any possible excess in the quantity of air admitted to the furnaces. But an indirect loss would still be experienced in the increased amount of heating surface necessary by reason of reduced efficiency of the surfaces for the abstraction of heat from the gases. A larger chimney is also required to carry off the waste gases when extensively diluted with air. Table II. gives the evaporative power of carbon, hydrogen, and good average coal. In the three first lines the figures refer to the several cases in which air is supplied in quantity accurately adjusted to chemical requirements. In the three lower lines the figures for coal are given on the assumption that air is supplied in excess to the extent of 30, 60, and 100 per cent. in the respective cases. In the first column the appropriate temperatures of furnace are given, and in the four remaining columns the respective weights of water evaporated from and at 212° F. In every case the figures refer to the ideal condition in which no heat

TABLE II.—FUEL, FURNACE TEMPERATURES, AND EVAPORATIVE POWERS, DISREGARDING ALL LOSSES.

Air and fuel admitted at 60° F.	Temperatures of gases in furnace. Fah.°	Weights of water in pounds evaporated, from and at 212° F.			
		Heat abstracted from waste gases until cooled to 750° F.	Heat abstracted to 400°.	Heat abstracted to 200°.	Heat abstracted from waste gases until cooled to original temperature of 60° F.
Carbon : air accurately adjusted to chemical requirements	5,040	13·08	14·18	14·76	15·18
Hydrogen ditto ditto	6,106	59·49	63·37	65·60	67·15
Average coal ditto ditto	4,746	11·49	12·49	13·07	13·47
ditto air 30 per cent. in excess	3,784	10·98	12·24	12·96	13·47
ditto 60 ditto	3,150	10·46	11·99	12·86	13·47
ditto 100 ditto	2,582	9·79	11·65	12·72	13·47

The above figures are based upon a calorific power of 14,652 for carbon, and 64,800 for hydrogen.

is lost by radiation or conduction to surrounding bodies, or by incomplete combustion.

Space necessary for combustion.—An ample amount of space over the fire should be provided for ensuring complete combustion. Its greatest service consists in the promotion of close contact between the unburnt gaseous fuel and the air, which arises from the eddying motion of the mixture. In Lançashire and Cornish boilers, the furnace tubes beyond the bridges act in continuation of the space over the fires if unobstructed, and chiefly for this reason it is unwise to place water tubes near the bridges. In marine boilers a combustion chamber is provided, which answers the same purpose If ample space is not allowed, heat is abstracted from the gases before the process of combustion is completed, combustion is impeded, and smoke is produced.

Boilers constructed with small tubes, through which the furnace gases pass, are especially liable to the premature cooling of the gases. On the other hand, the provision of an arch of fire-brick over the fire interferes very much with the withdrawal of heat from the gases, and thus promotes perfect combustion, and freedom from smoke production. Ample space over the fire possesses an additional advantage in reducing the liability to occasional overheating, which exists when the heating surface lies close over the fire. In gas firing, the air and fuel are so mixed as to secure almost instantaneous combustion, hence large space is not so imperatively necessary, but it is still very desirable in ordinary cases to obviate risk of overheating.

Analysis of gases necessary to show the amount of air supplied to furnaces.—If air were supplied in quantities exactly sufficient for the chemical requirements of the fuel, and if combustion were accurately completed, the furnace gases would consist of carbonic acid, water

vapour, and nitrogen, to the exclusion—practically complete—of all other bodies. Any excess of air admitted remains mixed with the burnt gases, but with its composition unchanged. The "analysis of the gases produced in combustion" is undertaken chiefly with a view to ascertain the amount of excess in which air is supplied. A second object is to ascertain the presence or quantity of carbonic oxide. Such analysis always receives careful attention in making boiler trials, and might be largely adopted with great advantage in ordinary current work. It should on no account be neglected in connection with any change in plant or attendants, and should be repeated at intervals for some time after a change takes effect.

Classes of fuel.—Coal, lignite, peat, wood, and coke, or charcoal from these, together with natural gas, liquid hydrocarbons, artificial fuel, and vegetable refuse are the chief materials used for heat generation. Coal may be broadly divided into two classes—bituminous coal and anthracite. The former contains large quantities of hydrocarbons, while the latter is further advanced in mineralogical change, and has lost much of its hydrogen. Anthracite generally occurs in proximity to rocks which have suffered extensive dislocation, so that the discharge of the volatile constituents of the coal has been much facilitated. Bituminous coal is necessary for the production of gas as usually supplied for illuminating purposes. Anthracite or coke may be used for the production of gas, of which the chief useful constituent is carbonic oxide. This is useful for heating purposes, but not for illumination, except when treated by the addition of other substances. Anthracite burns with very great heat, but is rather difficult to ignite. Coke and charcoal resemble anthracite very much in their composition, difficulty of ignition, freedom from smoke, production of great heat, and liability to burn away the

fire-bars, unless these are cooled by water or steam. Anthracite is very liable to crack into small splinters when first thrown on a hot fire, so that much fuel is lost by falling through the bars. It is often most successfully burnt upon a grate covered with from three to five inches thickness of loose clinkers, which prevent the burning of the grate bars. Bituminous coal is more easily ignited than anthracite, and burns freely, but without an excessive heat. It requires most unremitting care in use, so as to avoid the production of smoke. It is soft, and melts or cakes together on the fire, by which means the access of air is prevented, and the production of smoke further promoted. Almost the whole of the coal produced in the British Islands is of bituminous character. The largest field of anthracite occurs near Swansea. A very large proportion of the coal produced in the Cardiff district is semi-anthracitic or semi-bituminous, and is termed smokeless coal, which expression must be understood in a comparative sense. With the exercise of reasonable care in use, these coals need not, however, produce any serious amount of smoke, and on the whole they are the best steam coals obtainable. Lignite and wood are very little used in civilized countries as sources of heat for industrial purposes, but in their treatment the same principles apply as to coal, making allowance for the additional bulk.

Gaseous fuel.—When steam is required to be produced on a large scale by means of the combustion of refuse vegetable matter, the most advantageous course is to utilize it in the production of gas for firing boilers. Straw, cane refuse, and wood refuse are, however, often fired directly in boilers whose furnaces are constructed with special reference to the bulky nature of such fuel.

Estimation of calorific value of fuels from analysis.—The amount of heat to be obtained in the combustion of pure carbon and hydrogen have been already

given. That to be obtained from compound fuels of normal constitution can be approximately calculated from their analyses. This operation may be conducted according to Dulong's law, which is based upon the assumption that the whole of the oxygen present is already combined with hydrogen to form water, and that the remainder of the hydrogen and the whole of the carbon are so constituted that they will produce just as much heat as they would if supplied in a pure and uncombined condition. The results of such calculations are almost always corroborated within a small percentage by direct experiment with a fuel calorimeter, which should be resorted to in every case of importance. When fuel is carefully tested in a calorimeter, the results obtained should be adopted rather than the calculated ones, as a few coals and other fuels are so exceptionally constituted as to cause a variation of small amount, probably never exceeding 5 per cent. In the absence of a calorimeter, a knowledge of the analysis of a fuel is necessary in forming an estimate of its value. Usually the analysis is stated upon the assumption that the fuel is absolutely dry. When water is thus excluded from the analysis, it follows that all other percentages will be greater than when water is included. Analyses always give the proportions of weight. The following may be taken as the analysis of the good average coal, the calorific value of which is dealt with in Table II. in the condition in which it is usually received.

Carbon	75
Hydrogen	4·75
Oxygen	7·75
Nitrogen and sulphur ...	3·0
Ash	4·5
Water	5·0
	100·00

According to the usual method of calculation, one pound of this coal will produce 13,000 British thermal units of heat, or sufficient to evaporate 13·47 pounds of water, from and at 212° F. A very large proportion of the coal consumed in England answers fairly well to the above composition, with the exception that the ash may reach any figure up to 12 per cent., and occasionally exceed this amount, involving a corresponding reduction in the other proportions, and in the calorific value. The best Welsh steam coals in a pure and dry condition will furnish heat sufficient, if completely utilized, to evaporate 15½ to 16 pounds of water, or less in their ordinary commercial condition. Mineral oils produce more heat and wood produces less. The whole of this heat is first imparted to the furnace gases, and afterwards utilized more or less perfectly according to the conditions.

Sulphur, phosphorus, and other substances of a combustible nature enter into the composition of fuel, but their effect upon the calorific value is minute. Sulphur compounds derived from the fuel, and present in the furnace gases, are liable, in combination with moisture, to develop a corrosive action upon the plates of the boiler. Nitrogen exists in the form of ammonia, which is liberated on the application of heat, but without important effect upon the calorific value of the fuel. Oxygen has already been described as assumed to be in the condition of water, and also without influence upon the heating value. Water in the free state is almost invariably present in fuel, either originally or from exposure to the weather or to flooding. This varies largely under different conditions, but may generally be assumed to reach five per cent., which is not only a total loss directly, but causes a loss of heat in evaporating such water, and raising the vapour to the temperature of the gases.

Loss from ash in fuel.—The calorific value of nearly all fuels is materially affected by the amount of ash or incombustible matter which they contain. The ash contained in fresh vegetable matter is small in amount, that from dried wood being from one to two per cent. It is smallest in the large wood, greater in the branches and twigs, and greatest in the bark and leaves. Peat contains the same kind of ash as wood, and also, in addition, other mineral matter of an inorganic nature, such as sand, clay, lime compounds, &c., simply intermixed and derived from the soil upon which the peat has accumulated, or which have been deposited from suspension in water, with which it has been repeatedly overflowed during the period of its formation. Water rising from below is filtered, so that it brings no suspended matter along with it. But it may impart or remove soluble matter such as lime, according to circumstances. The ash of coal naturally contains the same elements as that from peat, and in addition mineral matters such as pyrites, which are deposited from solution in water, and subsequently changed, but which do not occur to an appreciable extent in peat. Coal has been found to contain from 1 to 12, and occasionally up to 20 per cent. of ash. Coals with the larger proportions are almost invariably uneconomical in use. An ordinary inspection is most unreliable as a basis for even the roughest estimate of the ash contained in a sample of coal, except as to the occurrence of gross impurity. As an extreme instance of this it may be remarked, that good cannel coal often contains large amounts of ash, sometimes 30 per cent. or more, though still valuable for special properties.

The proportion of ash contained in coke or charcoal is greater than that in the original fuel, for the reason that the volatile matter is removed, and consequently

the total weight is reduced while the ash remains. As an extreme case, the celebrated Bog-Head Cannel produces coke containing 75 per cent. of ash, which is quite useless as fuel.

Many coals which contain large proportions of ash are only eligible for use in situations where ready means are available for the disposal of the waste. Even in such cases the real value of the coal is proportionately low, and the cost of carriage and handling becomes important. A large proportion of ash is practically certain to cause a serious waste of good fuel, carried away in the ash, and often amounting to an addition of 50 to 100 per cent. to the matter absolutely incombustible. The loss of time, the disturbance of the steaming of the boiler, the loss of heat in the ashes removed, and the cooling of the boiler by the entry of cold air during the cleaning of fires, are also items of consequence. The whole question of ash economy is of such importance as to render an occasional test desirable where convenient means are available for weighing the waste. This would be most reliable if extended over a week, but a shorter period would suffice with care. In very few cases is it impossible to check the amount in some way. Care should be taken that the observations cover a definite time, with fires clean at the beginning and the end; and the ashes should be kept dry. In some cases, a quantity of ash is carried forward by the draft, and deposited in the flues or passed through the chimney into the open air. The absolute weight of this matter is, however, seldom important.

Care is necessary in sampling coal for analysis for ash, on account of the variation which exists in the ash from different parts of the same piece.

The amount of ash in coal of small sizes is often very much reduced by the adoption of a washing process.

CHAPTER III.

CALORIMETERS FOR PROVING THE VALUE OF FUEL.

Use of calorimeter.—Fuels of nominally the same class vary appreciably in their heating power. A knowledge of the actual heating power is obtained by means of the combustion of a certain weight of the fuel in a vessel, so arranged that the chief portion of the heat generated is imparted to a definite quantity of water. The observed increase in the temperature of the water furnishes the basis upon which the amount of heat developed may be calculated. This test is made upon simple fuels, such as carbon and hydrogen; upon compound solid fuels such as coal, coke, peat, and wood, and upon liquid and gaseous fuels.

Authorities differ very much as to the degree of confidence which should be accorded to Dulong's law, but there is no doubt as to its approximate truth when applied to solid fuels. Many details affect the absolute accuracy of application of any rule for this purpose, and the experimental verification of calorific values is a more simple, expeditious, and satisfactory process than the combination of chemical analysis and calculation which otherwise must be resorted to.

Essential parts of calorimeter.—The apparatus used in the determination of the calorific power of fuel is

called a calorimeter, which means a measurer of heat. As the term is applied to apparatus in which heat is measured for different purposes, it is well to distinguish the present class as "fuel calorimeters." The essential parts are—(*a*) An outer vessel containing the water to receive the heat. (*b*) An inner vessel of comparatively small size, to provide space for the operation of combustion. This is provided with means for the supply of oxygen or air, for the support of combustion; and, in the majority of cases, with means for the discharge of the products of combustion. (*c*) A thermometer for measuring with great exactitude the temperature of the water, before and after the operation. Accessories are also required for weighing the sample, for the supply of oxygen, and for firing the sample.

Use of oxygen to support combustion in calorimeter.—Oxygen is almost always used to support combustion, because it is more reliable as to effecting absolutely complete combustion; it occupies less space than air, and the products of combustion are more easily dealt with. The use of oxygen is quite appropriate, as it is well established that the amount of heat produced by the combustion of a given weight of fuel is precisely the same, whether effected by pure oxygen or by oxygen furnished by atmospheric air. There are, indeed, very good reasons for believing that any given weight of oxygen will generate the same amount of heat in its combustible union with any other substance, when the same is completely corrected, including correction for latent heat consumed or liberated.

Combustion in calorimeter under atmospheric pressure.—Until very recently, combustion has been effected in all calorimeters under atmospheric pressure, oxygen being supplied very steadily, so as to give the fullest opportunity for the whole of the heat produced to be

absorbed by the water. The operation should, however, not be so far prolonged as to allow any considerable loss of heat by radiation. But in the use of such apparatus, the extremely complicated character of the necessary corrections, including those on account of acid substances produced and dissolved, and the condensation of water-vapour produced, and also the existence of some uncertainty as to heat remaining in the products of combustion, notwithstanding all possible precautions, prevented the attainment of the desirable degree of reliability in the results obtained.

Combustion in calorimeter under high pressure.—Within the last few years M. Berthelot has devised a fuel calorimeter on new lines. He encloses the sample in a strong steel shell or "bomb," and supplies oxygen at a high pressure from a cylinder. The charge is fired by electric current, and combustion takes place with great rapidity. The whole of the products remain in the bomb, in the same condition as if combustion had occurred in the usual manner, so that no corrections are necessary on account of solution or of liberation of latent heat. A trifling amount of heat is mechanically absorbed by reason of the greater pressure in the bomb than before, but this is quite insignificant. The heat generated is quickly imparted to the water, and the temperature continuously noted until two or three minutes after the maximum is reached. The chief part of the heat generated is imparted to the water and to the bomb, the combined heat value of which is known. Some is absorbed by the different parts of the apparatus, and some is lost by radiation; the amount due to these two causes is assumed to be withdrawn from the water at a uniform rate. This rate is ascertained by observing the fall in temperature during one minute after the maximum is reached. The loss in

temperature thus obtained for one minute is multiplied by the number of minutes from the time of firing the charge to the attainment of maximum temperature. The product added to the observed maximum temperature gives that which would have been attained if the whole of the heat generated had remained in the bomb and the water without loss. This is the only correction required, and though the method of treatment is not absolutely above question, it is impossible that any error can arise in this way, to be at all compared with those arising in the use of any other form of calorimeter.

Reliability of results obtained by the Berthelot calorimeter.—The use of the Berthelot calorimeter has disclosed sensible inaccuracies in calorific powers, as previously obtained by means of other instruments. The most important of such corrections is that of pure carbon, which is usually taken at 14,400 British thermal units per pound of carbon, while the Berthelot has shown it to be 14,652 British thermal units. This correction of nearly 2 per cent. is of greater importance than is directly due to its magnitude, as it provides the means whereby the views and classifications of fuel adopted by different authorities may be much more nearly reconciled. The differences observed in the calorific powers of practical fuels are greater than in the case of pure carbon. In all cases the results are more uniform and reliable, and much more conveniently obtained. A sample of dry fuel can be proved by one man without assistance in a few minutes, and a less degree of manipulative skill is required than in the use of any previous apparatus.

Berthelot-Mahler calorimeter.—The original apparatus of M. Berthelot is very costly, chiefly owing to the quantity of platinum used in its construction. M. Mahler has modified it with a view to largely reduce

the cost. No sacrifice in efficiency is involved, but, on the contrary, the proportions of the instrument are such as to render it conveniently applicable to the proof of gaseous fuel if required. The apparatus is now made in England by Messrs. Bryan Donkin and Co.

Extended use of calorimeter is desirable.—Accurate and satisfactory treatment of questions effecting the consumption of fuel is quite impossible in the absence of a definite knowledge of its calorific power, and many firms would derive great advantage from frequent use of the process. But probably in the majority of cases, even its present reduced cost will prevent its direct application. There is, however, no good reason why the work should not be much more extensively performed by analysts for a moderate fee.

Selection of samples.—In selecting a sample, care should be taken to obtain one which is representative of the whole. Pieces of average quality should be selected over the surface of the heap, and where possible from several days' deliveries. These should be broken down to the size of wheat, well mixed together, and a sample taken without sifting. If the whole is reduced to fine powder, combustion takes place with explosive violence, unless air be substituted for oxygen. If necessary the sample may be dried by exposure to a gentle heat for 12 hours. If during this process the temperature is allowed to exceed 100° F., some of the hydrocarbons contained in the sample are likely to be lost by distillation, and so affect the result.

Separate determination of ash.—When a sample is proved in the calorimeter, the ash is dispersed all over the combustion chamber, and cannot be collected for weighing without considerable difficulty. The importance of ash estimation is, however, less in this process

than in calculation of calorific power from the analysis. If required, the amount of ash may be determined by firing a second charge much more slowly than the primary one. This object may be secured by forming the sample into pellets, or by the use of air, with or without the addition of oxygen.

References.—Reference may be made to the *Journal of the Iron and Steel Institute*, 1892, Part I., for a paper by Mr. Thwaite, giving accounts of the Berthelot-Mahler and the Thomson calorimeters; also to the *Minutes of Proceedings of the Institution of Civil Engineers*, vol. cviii. pp. 457, 459, for notes on the Berthelot-Mahler instrument. A *Treatise on Fuel*, by Professor Galloway, gives accounts of the Ure and the Thompson calorimeters, and the one of Favre and Silberman as modified by Kestner and Meunier. This work also contains much useful information on calorimetry.

CHAPTER IV.

STORAGE AND MANIPULATION OF COAL.

Drying of fuel.—Solid fuel, stored in a dry cool place for a moderate length of time, usually loses any excess of moisture which it may contain, whereby its condition for use is improved.

Loss of volatile constituents of coal.—Coal also usually contains, absorbed in its pores, a quantity of mixed gas which is in a condition of instability. In the mine, this gas is often discharged with great force, showing that it must exist condensed under great pressure. At other times it is given off comparatively slowly, and whether in the mine or when stored above ground, in the absence of suitable means of ventilation, the gas is very liable to form an explosive mixture with air. This gas is a mixture in which marsh gas predominates. It is often given off at atmospheric pressure in quantity sufficient to occupy, in a gaseous form, much more space than the coal from which it is derived. This action shows the porous nature of the coal, and bears upon the liability to heating, and to still further loss during storage.

Heating of coal by absorption of oxygen.—The oxygen of the air is absorbed into the porous mass of the coal, and in the process causes an increase in temperature. In this absorbed condition, oxygen and any other gas

which the coal may happen to contain possess very active chemical properties, so that they are particularly apt to combine with each other, or with the constituents of the coal, and, in the act, to cause a further increase in temperature. This heated condition, though apparently very slight, often leads to a slow combustion, which causes a considerable reduction in the weight and calorific value of the coal. The most volatile and easily combustible portions are first attacked, leaving the more intractable matter and the ash behind. This action is most prominently developed when the material is reduced to a condition of fine division or dust. All forms of carbon used as fuel are apt to exhibit it, but lampblack is pre-eminently liable to take fire if stored in quantity, and to set fire to any adjoining woodwork. Powdered charcoal also shows this action. Professor Lewes says that 100 bushels placed in a heap will always ignite. Anthracite coal possesses this property in a very small degree, and other coals in a greater degree. The action appears to be independent of the presence of slight moisture, but probably any important amount of free water would retard or prevent it. A local application of water to a heated mass sometimes causes a sealing action, which prevents free access of water to the fire, so that its extinction by means of water is difficult. Such fire may, however, be extinguished by the use of a copious and well-directed supply of steam.

Heating by oxidation of pyrites.—Coal is usually accompanied by iron pyrites, the sulphur and iron of which also possess strong chemical affinity for oxygen. The absorption of oxygen by pyrites causes the mass to break up into small pieces, or into powder, raises the temperature, and promotes the further absorption of -ygen, both by the pyrites and by the pure coal. This · proceed very slowly, or it may extend with great

rapidity, according to circumstances, and is often accompanied by a sublimation of sulphur. The heating due to the presence of pyrites is shown to be greatly promoted by the presence of water, though probably not in all cases actually due to it.

Danger caused by presence of pyrites.—Pyrites has been shown by experience, both in the mine and above ground, to be the agent which usually, though not invariably, originates heating. For this to take place it is not necessary that the total amount or percentage of pyrites should be singularly large. If in one piece only there is a considerable amount of pyrites, this is quite able to act as "kindling" to set up the action, which will then more easily spread over the whole mass, and receive support from the hydrogenous constituents of the coal.

Precautions to be observed in storage of coal.—Firing of the mass may usually be prevented by avoiding storage to a great depth, or by otherwise securing free access of air to cause the prompt dissipation of any heat which may be generated. If this is delayed until the mass becomes much heated, the admission of air is very likely to increase the action. If, however, air can from the first be completely excluded, heating will be prevented with certainty, but air which is enclosed with the coal when shut up may form an explosive mixture with gas evolved from the coal.

Storage of fine dust or slack should be avoided as far as possible. Where mixed coals are received for storage, an attempt should be made to use the fine stuff at once and store the large only. But if this cannot be done, the small should be distributed in layers, and not placed together in mass. All remainders from previous storage should be scrupulously cleared out before the storage of fresh coal is commenced. Coal should be examined

when placed in store, and all pieces which appear to contain pyrites removed, or at least exposed to the air on the top of the heap.

Degree of combustion.—Actual firing of coal occurs very seldom. But a change which amounts to a slow burning of the best part of the coal often happens, to an extent quite unsuspected. Gas coals which contain large amounts of volatile matter are especially liable to heating and waste in this manner. In cases where the amount of wasting of gas coal has been ascertained, it has proved to reach one-half of the total heating value in six months. Pure anthracitic coals which contain little volatile matter or pyrites are practically free from tendency to waste. Between these extremes there is an infinite number of gradations.

Arrangement of coal store.—A coal store should be roofed over to exclude rain, but the sides should be as open as possible to admit a free circulation of air. Temporary covers of corrugated iron or boards arranged to shed the rain may be used on an emergency, secured against stripping by wind. No wood should be used in any part of the construction so disposed as to come into contact with the coal. The floor should be sunk so that in case of fire it may be possible to use water effectively for extinction. The surface should be smooth for shovelling over, for which purpose concrete properly laid is excellent. Owing to the effect of heat in promoting decomposition, special care should be taken to avoid placing coal in proximity to flues, chimneys, steam-pipes, or other sources of heat.

Cost involved in storage.—A quantity of coal sufficiently large to avoid serious risk of interruptions to work on account of stoppage of coal supply should be provided, but this should not be exceeded on account of the possible loss in storage. The cost of storage includes

the cost of ground and erections, the interest upon the cost of coal, the cost of labour working into and out of store, the risk, and the depreciation of the coal itself. The money value of the loss arising from interruptions of work varies very widely in different cases. But in such undertakings as gas-works, railways, water-works, pumping stations, &c., the idea of suspension of work cannot be entertained. When coal is received by water, two or three weeks' supply should always be held in winter.

Labour-saving appliances for storage of coal.—In some instances the conditions admit of the adoption of measures whereby manual labour in trimming coal is dispensed with. This object is fully secured when the coal can be received by railway truck at a level much higher than that of the stoke-hole floor, but always involves the production of large quantities of dust in dry weather. Coal may be obtained in trucks with drop bottoms, or with falling doors at the side. The best drop-bottom wagons will discharge almost every ounce of the load, but side doors leave a large quantity to be shovelled out, or afterwards dropped through the bottom. If the bunker is very little wider than a wagon there is no object in receiving coal by other than drop-bottom wagons. But if the bunker is of considerable width, the use of side doors will give the means for storing a much greater quantity of coal, and for reducing the distance it will fall from the truck, and consequently the amount of breakage and of dust raised in the operation. In some rare instances it may be convenient to receive coal in trucks with end doors. These should be brought in upon rails of which the last 10 feet fall about 3 feet, to give a sufficient slope to the floor of the truck, so that the coal will run freely. In these, as in all cases, measures should be adopted to

prevent accidental derailment of trucks. Platforms should also be provided to allow safe and convenient operation of the doors.

Construction of coal bunkers.—The walls of a coal bunker should be of brickwork. Boards may be used if lined with sheet iron, but they are less durable and more unsightly. They also allow the passage of dust. The strength of the walls should be as great as would be required to resist the pressure if the bunkers were full of water. Tie-rods are of great value in supporting the walls against high pressures, but should be arranged to avoid interference with falling coal.

Anti-breakage plates are sometimes used to check the fall of coal. These are costly to provide of adequate strength, and they cause a sensible reduction in the storage capacity, while the purpose with which they are provided can usually be effected by care in discharging the coal.

The floor of the bunker should be made of, or covered with, iron plates, and should be set at such an angle as will cause the coal to run steadily. This is secured when the angle of the floor is $33\frac{1}{2}°$, which is equivalent to a rise of 1 part in a length of $1\frac{1}{2}$ parts. The floor will be required to be made of great strength to support the whole of the weight in the bunker, and tied securely together so as to resist any wedging action which may arise from the angular position of the floor. A small piece of floor adjoining the door by which coal is withdrawn from the bunker may be made level, or nearly so, with a view to increase the quantity of coal stored. When this is the case, the door will require to be larger than would be otherwise necessary. In all cases the doors provided to close openings in walls should be arranged to slide easily, and to adjust by chain and hook.

Coal-conveying appliances.—In some cases travelling bands are employed for conveying coal a considerable distance from bunker to stoke-hole. The door for allowing the coal to escape from the bunker is then placed beneath the floor, and arranged for fine adjustment. The door must move in a direction parallel to the band, so as to secure at all times a fairly uniform distribution upon the band. The coal may be delivered from the band at any part of its length, but must all be delivered at the same place at one time. Worm conveyors are also used for similar purposes, and for the supply of coal to mechanical stokers, for which purpose they are particularly adapted for the reason that they will discharge coal simultaneously at several points.

Coal-raising appliances.—Coal is sometimes raised by means of an elevator on the band and cup principle. It may be fed to the elevator either directly from the bunker, or by means of a travelling band or a worm conveyor. At Messrs. Fielden's mills at Todmorden the coal is received in hoppers from carts, fed to an elevator by means of a worm conveyor, and carried from the elevator to the several stokers by means of a second worm conveyor.

CHAPTER V.

COAL WASHING FOR THE REMOVAL OF SOLID WASTE.

COAL exists in the mine in contact with various substances which possess no combustible value. These are chiefly shale, iron pyrites, and the underclay which is the exhausted soil in which flourished the plants whose remains have been transformed into coal. The impurities imparted to coal in the several ways vary in proportion in different cases. When pure coal lies in thick beds uninterrupted by seams of impurity, and when the roof is of stone, and the underclay dry and hard, the coal is brought to the surface with practically no admixture of impurity except the organic ash which was originally secreted by the vegetation in its growth. This may form from 1 to 5 per cent. of the whole, and being uniformly diffused throughout the substance of the coal, cannot be removed by any process which does not involve its complete destruction. A large proportion of the coal raised unfortunately contains much more impurity than that just described. In average practice, the proportion of shale and pyrites raised in the coal depends very much upon the strictness or laxity of supervision exer-

cised over the workmen in the mine. The majority of the larger pieces are also removed by hand picking at the screens. But the smaller pieces of these and the whole of the clay are practically incapable of removal in this way. By screening for size, these impurities are removed from the large coal, but only to accumulate in the small coal, so that the percentage of impurity in the smallest coal may be several times greater than the average proportion of impurity in the mixed coal as raised from the pit. Small coal realizes in the market a lower price than large coal from the same bed, chiefly on account of its inferior condition as to purity. But when coal of any kind, intended for ordinary consumption, contains more than 10 per cent. of impurity, steps should be taken with a view to its reduction.

Washing is the only process which is really effective in the removal of impurities. The coal is not in this sense subjected to a simple surface washing with a view to improve its appearance; but the process is really one of sorting or separation depending upon the facility with which nodules or particles of solid matter are deposited from suspension in water under different conditions. For such a purpose the water may be still, but is more frequently in motion, either continuously or intermittently. Assuming that a collection of bodies of equal size, but of different densities, are placed in water, those of the greatest density will become first separated, after which those of medium density, and lastly, those of low density. This will occur whether the water be still or moving. A certain depth of water, when set in upward motion, becomes equivalent to a greater depth of still water, while the action is further improved by imparting pulsations to the water. The several densities

of material may also be separated by placing the mixed collection in water which is directed along a channel at a suitable velocity. The pieces of greatest density will be first separated, and those of lowest density will remain suspended the longest time. In like manner, a collection of bodies of equal density, but of different sizes, may be separated, the largest becoming first separated and the smallest being carried the greatest distance. The latter action is utilized in continental practice for the sorting or sizing of coal which is too tender for suitable treatment on screens; but as affecting English practice, the action is chiefly important as an element of complication, owing to the tendency of small coal to deposit along with impurities of a still smaller size.

Coal is of less density or specific gravity than any of the impurities usually associated with it, clay being the nearest, the specific gravity of the former being 1·28, and that of the latter 1·92. The comparative weights in water are, however, obtained by the subtraction of 1 from the specific gravity. Thus, in water, one cubic inch of clay weighs more than three times as much as one cubic inch of coal, and separation takes place with greater facility than would be indicated by the relative specific gravity. Proposals have been made to take still greater advantage of this principle by the use of liquids of greater density than that of water, but these have not met with much favour. The accurate separation and sizing at one operation of coal which contains much small, is, however, found to be impracticable. In such cases it is found best to perform the first washing with a view only to the removal of the smallest clean coal, the remainder being afterwards operated upon either once or twice. The variations which exist between

different coals, as to size, impurity, hardness to endure treatment, and the value attached in the market to purity, render necessary special treatment in almost every case. Fan-blast has also been adopted in some cases for the separation of fine dust.

If it were practically and commercially possible to carry the sizing of coal by screens to a much further extent than is now practised, the separation of refuse would be correspondingly facilitated and better results secured. Under present conditions it is impossible to avoid some appreciable waste of good coal, though often less than would be lost in the ash of unwashed coal.

Much coke of excellent quality is made from coal which would be unsuitable for such purpose without washing. The improvement thus secured chiefly consists in the great reduction in sulphur which is effected by the removal of pyrites. For the same reason a valuable improvement is effected in gas coal. In steam coal the removal of sulphur is a distinct advantage, but in a commercial sense the removal of the incombustible constituents which form clinker is more important. These substances increase the cost of transit and handling at every point, and cause great difficulty in their ultimate disposal. The practice has become a necessity at many coal-mines on the Continent, where the small thickness of the seams and the amount of impurity in the coal are much more serious than is generally the case in England. In England, however, the advantages secured are becoming more clearly realized by consumers, who receive much better value for the outlay, so long as obvious precautions are taken against the retention of an excessive amount of water in the washed coal.

Reference may be made to the *Minutes of Proceedings of the Institution of Civil Engineers*, vol. lv. p. 259, and vol. lxx. p. 106 ; also to *Engineering*, vol. xxix., and to the *Transactions of the Mining Institution of Scotland*, "Report of Coal Cleaning Committee."

CHAPTER VI.

CONVECTION, CIRCULATION, EVAPORATION, AND PRIMING IN BOILERS.

Circulation of water depends upon expansion by heat. —The expansion of liquids by means of heat is a phenomenon of very great importance in connection with the working of steam-boilers, for the reason that it causes convection or circulation, by which means the transfer of heat from the hot gases in the tubes and flues to the water in the boiler is very much facilitated.

Experiments.—If a test tube is nearly filled with water, and heat applied at a point a little distance below the surface of the water, the water will very quickly boil above the point of application of heat, but on account of its low conducting power for heat, the portion below the point of application of heat will rise in temperature very slowly indeed. If the experiment be afterwards repeated, but the heat applied near to the bottom of the test tube, the whole of the water will boil almost as rapidly as the upper part did in the first experiment. The difference observed is due to the increased circulation in the latter case arising from the expansion caused by heat, which renders the heated particles of water lighter than before, causing them to

rise and to be replaced by the colder particles. This is shown again more clearly in a thin glass flask of water heated from below. The water above the flame rises, and a constant circulation takes place throughout the mass. This is seen with special distinctness when the flame is a little towards one side, and when a small quantity of bran or other light material is added.

Deductions from experiments.—These experiments show the conditions prevailing inside a steam boiler when it is at work. The furnace gases impart heat to the plates, which again transmit it to the water, which carries it away at once. The circulation thus caused tends to effect a uniform rise in temperature throughout the water. But this can only occur when there is water of a lower temperature—and therefore heavier—so situated as to readily take the place of the lighter water as it rises. In the case of an ordinary marine boiler with two furnaces, when raising steam from cold water, the heated gases from the burning fuel are almost entirely confined above the level of the grate bars, and the water becomes heated all around the furnaces and the combustion chambers to one dead level. Such a condition is precisely equivalent to that of the first experiment with the test tube, in which heat was applied near to the top of the tube, and in which the lower part of the tube received heat very slowly. With very little qualification, the same conditions apply to marine boilers with three or four furnaces. Though Lancashire and Cornish boilers are more fortunately situated, by reason of the application of hot gases to the lower part of the shell, the advantage due to this fact often only comes into force after a lapse of some hours from the time of lighting fires, on account of the very great cooling effect upon the gases exercised by the cold boiler and its contents. Vertical water tubes in

the furnace tubes will cause a circulation, if the water within them is heated more completely than that outside the furnace tubes. But in raising steam it often happens that the gases in the furnace tubes heat only the upper part of the water tubes, and thus fail to set up any circulation at all, until after such a lapse of time that a circulation from below is developed by reason of the hot gases beneath the boiler. Defective circulation leads to straining and leakage of boiler, to loss in steaming power and economy, and to internal corrosion of the structure of the boiler.

Increase in activity at boiling point.—The experiment with the glass flask shows the upward and downward currents which are developed on the application of heat, and the gradual way in which they gain strength as the temperature rises, until when the water boils they suddenly become exceedingly active. The bubbles of steam which are formed when the water boils reduce the total density of the water very much, and account for the sudden increase in the power of the current at that time. Steam of high pressure occupies less room than steam of low pressure, and the bubbles are then of correspondingly smaller size, causing the circulation to be steadier.

Provision for space to allow circulation.—It is evident that for the free transmission of heat from the heated plates of the boiler to the water contained within it, every facility must be given for an efficient circulation, both upward and downward. This point is often overlooked or under-estimated in connection with boilers, and especially multitubular ones. In some cases the boiler is quite crowded with tubes, leaving very small space for the upward currents, and totally inadequate space for the downward currents along each side, so that an efficient circulation becomes quite impossible. Under

these conditions, the water in the boiler fails to become uniformly heated, excessive priming occurs, also leakage at the tube plates, and the wear and tear is far greater than there is any necessity for. Many boilers in this condition are over-tubed to such an extent that the draft through the tubes is too feeble to keep them clean, they become stopped with ashes, and thus become worse than useless. In good marine practice, the spaces between and outside of the nests of tubes are large enough to admit a man, for examination and cleaning, which usually gives space sufficient for circulation when in work. In many other cases the practice might be followed with advantage. The spaces between the several tubes are also often far too small.

Circulation in a vertical boiler.—In vertical boilers of all sizes the shell is usually parallel. In the majority of cases the fire-box is also parallel. But a fire-box of tapered form, giving a wider water space at the top than at the bottom, is much better. This facilitates circulation, and a boiler fitted in this way is always much appreciated by crane drivers on account of its free steaming properties.

Use of separating plates.—Most engineers in charge of boilers have at some time felt a desire to improve the circulation of a troublesome boiler by the use of plates to separate the up and down currents, or to strengthen the up current, by confining it in more close contact with the heating surface. The consideration of the subject generally ends in the discovery, that the size of the manhole to admit plates, and other difficulties, prevent the application of such means. In the comparatively few cases in which they are applied, they cause trouble in cleaning the boiler, and they are seldom so successful as might be anticipated, one important reason for which being the one already

referred to, which sometimes interferes with—or prevents—circulation in vertical water tubes when raising steam, at which time appliances for promoting circulation are required with the greatest urgency.

Amplitude of space.—As affecting the time during which boilers are in ordinary work, there are no automatic and self-contained means for ensuring good circulation which can compare with the provision of ample space against all heating surfaces, for the free development of the up and down currents.

Circulation as affected by manner of setting boiler.—In Lancashire and Cornish boilers, the gases make three passages along the length of the boiler. The first is through the furnace tubes, the second is below the boiler usually from back to front, and the third is a divided one, from front to back along each side. The gases in the second run are of higher temperature than those in the third run, and the heat is applied to best advantage at the bottom of the shell, while the comparatively cool gases in the third run, which are applied to the sides, do not impart sufficient heat to the water to interfere seriously with the downward currents against each side. Boilers of these types, therefore, possess a very efficient circulation when in work. As elsewhere explained, boilers which are not heavily worked are sometimes set with the second and third runs exchanged.

Amount of heat imparted under different conditions.—Of the entire amount of heat generated in the furnace, a very large proportion is at once imparted to the heating surfaces exposed to direct radiation from the fire. In a vertical boiler with a plain fire-box, in which the whole of the heating effect is produced in this way, the amount of water evaporated may reach from one-third to one-half of that evaporated per pound of fuel

in a thoroughly efficient boiler of the Lancashire type. For this, and for other reasons suggested by the results of trials of locomotive boilers divided into sections, it would appear that from 20 to 25 per cent. of the whole heating power of a Lancashire boiler is exercised immediately over the fires and bridges. The remainder of the furnace tubes contribute very largely to the total, but on a descending scale as the back end is approached. The heat thus applied is almost entirely expended in converting into steam the water which has been warmed by the heating surface of the shell. Probably some steam is raised upon the heating surface at the bottom of the boiler, above the central flue, which steam will rise between the furnace tubes and assist the general circulation. The heating surface at the sides is chiefly effective in raising the temperature of the water, especially when this has not been subjected to preliminary heating in an economizer. The equivalent amount of water evaporated per square foot of total heating surface of the boiler is usually from 5 to 7 pounds in economical work, though more or less may be obtained if required. Similarly the amount evaporated per square foot of heating surface over the fires and bridges is probably 30 to 42 pounds per hour. In each case the figures are for the sake of uniformity based upon the assumption that the water is supplied at a temperature of 212° F. and evaporated at that temperature. The amount of water converted into steam even in the hottest part of the boiler is never so great as to occupy a very large volume. Even with an evaporation of 42 pounds per hour, at a pressure of 60 pounds per square inch, the volume would be only 4 cubic feet per minute, which would not appear to be likely to lead to any difficulty. But it is found that in efficient work a constant stream of water must flow

over the heating surface, and carry with it the particles of steam as they are formed. Otherwise these particles retard or interrupt the passage of heat to the water, the heat accumulates in the metal, and when this occurs in the parts over the fires, the metal is quickly destroyed.

Results of variation in temperature due to defective circulation.—Boilers of nearly all types in common use are defective in circulation when raising steam from cold water, and are in consequence exposed to stresses of a very serious character, arising from the difference in temperature between the upper and the lower parts. If a boiler in this condition were to be cut along the line of separation between the hot and cold parts, the two would be found to be of very different length; but they must assume the same length when they are connected together, consequently the two parts exert very great strain upon each other. It is practically impossible to avoid this straining action, except by the adoption of such measures as will ensure greater uniformity of temperature throughout the boiler.

Equalization of temperature.—An excellent means for equalizing the temperature consists in supplying the boiler with hot water. When an economizer is used, and other boilers are at work, this may be done by pumping the water into the boiler through the economizer, when it will be supplied at boiling temperature. The steam arising from this heats the boiler with a fair degree of uniformity from the first, and no objection need be raised; during the operation, however, the air in the boiler should be allowed to escape. Steam may also be applied at the bottom of a boiler filled with cold water. In this way a jet is used, so directed as to induce a current and circulate the water throughout the boiler, while its temperature is increased by the addition of

the steam. When this is done the boiler should be quite free from loose dirt, which would be liable to be picked up by the circulation and deposited upon the furnaces, where it would give trouble by causing overheating. A donkey pump may also be used to draw water from the bottom of the boiler and return it through the ordinary feed valve. If this is done while steam is being raised, the entire mass of water will uniformly increase in temperature, but for this purpose a very small donkey pump should be avoided. In many cases the adoption of the jet or donkey pump would involve the provision of a small special boiler to supply steam. The cost of this would, however, be fully repaid by the result if the main boiler is seriously deficient in circulation, which leads to leakage and very great wear and tear. The donkey pump is especially useful in exciting circulation if required after a stoppage over-night or at week ends.

Uniform circulation in coil boilers.—Many types of tubular boilers are made, in which ordinary convection does not take effect. In these the water is admitted into one end of a tube, or into a vessel communicating with a series of tubes, and is withdrawn at the other end as steam. Probably in all cases, local convection on a very small scale is developed. Usually, the several parts are designed so that expansion and contraction may freely take place, according to the temperature of the whole or the several parts. There is no possibility of the occurrence of a line of considerable length passing through continuous metal, and along which the metal is exposed to a high temperature on the one side and a low one on the other side.

Corrosion arising from defective circulation.—Corrosion of the internal surfaces of a boiler is found to occur largely in parts of the water space in which circulation

is defective. This is more pronounced in situations exposed to high temperatures, but is far from being confined to these. Possibly this arises in some way from the air or oxygen in the water, but the exact nature of the action, chemical or physical, has not been determined. The fact is, however, only too well established, and furnishes an additional reason for taking every precaution to ensure good circulation.

Spasmodic production of steam.—Under ordinary conditions, heat applied by means of a heating surface to water which is already raised to boiling temperature, causes the transformation into steam of some of the particles in contact with the heating surface. This steam accumulates, forming bubbles which rise to the surface and collect in the steam space. Under certain conditions, the heating surface appears to possess a power of attraction over the water, so that the bubbles fail to be produced in a continuous stream, but are produced in a spasmodic manner throughout the mass of liquid. If the water is very limpid, the steam will unite to produce very large bubbles. But if the fluidity of the water is impaired by the presence of any foreign matter, the steam is prevented from coalescing into bubbles, and remains diffused in the mass. In either case, if the water is not contained in a closed vessel, it will be forcibly ejected, in the one case as a liquid, violently lifted out of the vessel by the bubbles of steam, accompanied by noise and shock; in the other case as a frothy mass, approximately uniform in character, and without the accompaniment of shock or considerable noise. These actions are well shown in a glass vessel with a clean surface, but they are prevented or largely modified by the occurrence of small objects or particles of metallic nature, either lying loosely in the vessel, or attached to the glass. Glass or metallic

vessels with varnished surfaces in contact with water cause a more powerful action than clean glass vessels. Clean metallic vessels give rise to similar action in a less powerful degree than either of those described. Varnish and other similar substances appear to have a molecular attraction for water, and the same applies to the heating surfaces of some boilers. The precise condition of surface which has this effect is not accurately defined. But it is well established that such action, which originates in the character of the surface, is promoted by the presence of soda and other substances in solution or suspension, or the presence of oil or soap diffused in the water. Water which has been freed from dissolved air, either by slowly heating to the boiling point in an open vessel, or by freezing, is particularly subject to such action. Any reason which interferes with the uniformity of the mass, as, for instance, a sudden change in the class of water fed into a locomotive or marine boiler, will cause irregular ebullition.

The irregularity in ebullition just described arises by reason of the fact that water in the conditions referred to, may be raised to a temperature above that at which it usually boils, but that when such point is reached, the temperature suddenly falls to the temperature corresponding to the pressure at the time applied to the liquid, and a flash of steam is formed. A difference of this kind, amounting to more than 60° F., has been produced by the adoption of special precautions, and the use of special apparatus; but a difference of 10° F. may arise accidentally. The latter difference is sufficient to cause one per cent. of the mass of water to instantaneously assume the form of steam, such change taking place throughout the mass. If in such a case the water is subjected to atmospheric pressure, the amount of steam produced will occupy seventeen times

as much space as the entire bulk of the liquid occupied previously to the change. In this way the whole mass of the steam and water is dispersed, and the vessel is almost if not completely emptied.

Irregularity due to variations in temperature and pressure.—In a boiler under pressure, the action is under much more effective control. Under a pressure of 50 pounds, a rise and fall of 10° F. in temperature will cause a production of steam occupying about four times the volume of water; and at 150 pounds pressure the volume of steam so produced is less than that of the water. But though the irregularity is much reduced under high pressures, it is in comparatively few instances that it is entirely prevented. Such action is known as priming, and sometimes causes the boiler to lose its water with great rapidity. During violent priming the water and steam in the glass tube of the water-gauge become quite confused, so that the actual level of water in the boiler is most uncertain. At such times the water carried along with the steam causes great risk of damage to the engine, by reason of accumulation in the cylinders, and causes damage to the scraped surfaces by reason of mud carried over from the boiler. A similar action is also produced by alternate changes in pressure. If a boiler is assumed to be working with uniformity at a certain pressure, and a sudden demand is made upon it for a quantity of steam largely in excess of its current production, such demand can only be met by drawing upon the steam contained in the steam space of the boiler. This again involves a definite reduction of pressure in the boiler. If this is done abruptly, to any appreciable extent, steam will be produced as by a sudden pulsation in temperature, and priming will occur. Under such conditions the production of steam, free from large quantities of unevap-

orated water, is quite hopeless. This condition is most clearly marked when the steam space in the boiler is small. A good illustration of this action, which is conveniently arranged, is furnished by a number of bottles of gaseous soda-water, stopped by corks, and opened with various degrees of rapidity. When this is done very slowly, the amount of effervescence is quite trifling. But in the cases in which the corks are drawn suddenly, a large amount of the contents will froth over the neck of the bottle. It will be noticed that the gas which gives rise to this action is produced throughout the mass of the contents. This experiment will probably fail if attempted with glass-stoppered bottles.

The abnormal evaporation which arises from occasional excessive demands for steam, is sometimes imitated, by reason of an engine of large steam consumption and slow speed of revolution drawing steam in a fitful manner.

Use of steam receivers on boilers.—Steam drums or receivers, generally of a cylindrical form, are often provided in marine boilers, and occasionally in land boilers, with the object of providing dry steam. This is possible by reason of the equalization of the current of steam from the boiler, whereby the pressure of steam in the main steam space becomes nearly constant, and the rate of ebullition uniform instead of variable. For the efficient fulfilment of these conditions, the volume of steam in the receiver must be sufficiently large and the openings from the boiler to the receiver sufficiently small. The velocity of the current of steam through these necks or openings should not be less than from 120 to 200 feet per second. Each neck should also communicate with a collecting pipe. The possibility of securing any advantage by the use of a receiver in ordinary cases may be somewhat questionable. But it

is impossible that one which fails to fulfil the conditions described can perform any good service. It is scarcely necessary to discuss the utility or otherwise of the small domes which were usually provided on boilers 20 to 30 years ago, but practically the same principles apply, with the addition that the large openings at the base of such domes caused such a loss of strength as quite forbids their use in the same way with the high pressures now prevailing.

Production of dry steam in boiler.—In a boiler which is free from very great malformation, true priming is most effectually prevented by the use of clean pure water, by frequent and careful cleaning of the whole of the interior, and by the avoidance of fluctuations in pressure. In ordinary stationary boilers, the adoption of high pressures and more careful attention to their operation has obviated serious trouble during recent years. But much trouble is still caused in connection with the lower pressures and irregular work to which small boilers are exposed; though not the most important, these are very numerous.

Collection of steam in boiler.—By means of priming the steam supplied by the boiler is contaminated by unevaporated water, which detracts most seriously from its value for use in an engine. But steam which is produced in an absolutely dry condition becomes similarly contaminated if allowed to flow rapidly past a surface of water, especially if the latter should be in a disturbed condition; a damp surface will have the same effect. This is not true priming, but for the sake of convenience, all unevaporated water contained in steam is termed "priming water," and the means adopted with a view to its prevention are termed "anti-priming" arrangements. The expedient most usually and successfully adopted for preventing the acquisition of water by

steam subsequent to its formation is a pipe of some considerable length perforated throughout, so as to collect steam at many points, and obviate the transit of a current of steam over the water surface. Strictly, the apertures should be so disposed as to collect the steam most freely at the part of the boiler in which it is most freely produced, which is over the furnaces, but this is not imperative.

Estimation of moisture in steam.—The most usual and reliable means for the measurement of the proportion of unevaporated water which may be present in steam acts by condensation. A vessel is charged with a known weight of water at atmospheric temperature, after which a certain weight of steam is added and the increase in temperature is noted. Assuming that 100 pounds of water are supplied at the atmospheric temperature of say 60° F., and that 5 pounds of steam are added, the boiler pressure of which is 150 pounds above the atmosphere, the heat contributed by the liquid water, calculated from zero F., equals $100 \times 60 = 6000$. That contributed by dry steam equals $1225{\cdot}45 \times 5 = 6{,}127{\cdot}25$. The total amounts to 12,127·25, which divided by the weight 105 pounds, gives 115·4° units of heat contained in each pound of the water, the increase in temperature being 55·3° F. If any appreciable proportion of the 5 pounds added consists of unevaporated water, such will fail to contribute liberated latent heat to the mixture, and the temperature will increase by a less amount. If in the above case 5 per cent. of the added quantity consists of unevaporated water the calculation will be as follows:—

100·0	lbs.	× 60	=	6000
4·75	„	× 1225·45	=	5820·88
·25	„	× 370·27	=	92·57
105·00		× 113·5	=	11,913·45

The increase in heat contained in one pound is then $113.5^{\circ} - 60^{\circ} \doteq 53.5$ units, and the temperature 53.4° F., or 3·4 per cent. less than before. Some heat is absorbed by the vessel which affects the result. For this reason a thin sheet metal vessel supported by an outer one of wood or other material of low conductivity is essential. In this respect thin zinc answers very well. It is scarcely necessary to say, that the weight and temperature of water at each stage of the operation must be observed with great accuracy.

An instrument has been lately devised by Professor Peabody, and made by Messrs. Schäffer and Budenberg, for the direct estimation of small quantities of water in steam. The action depends upon the fact that when high pressure steam is discharged through a nozzle without performing any work, its temperature is greater than that of saturated steam at atmospheric pressure. The difference of temperature thus arising is rather less when the steam is wet than when it is dry, the boiler pressure in each case being the same. In this instrument the steam is directed through a nozzle so as to impinge upon a pocket containing a thermometer. The readings from this thermometer are applied to a table or diagram, and the amount of moisture directly ascertained. This instrument is extremely convenient in application. It will show the presence of water in any quantity. If this quantity does not exceed 2·5 per cent. it also indicates the exact proportion, but not beyond that amount.

Other instruments are made which act by the separation of liquid water and its subsequent measurement. These, however, are more costly and less convenient for general application than those described. Such instruments are often called calorimeters, which is confusing and misleading, as the term is applied to instruments

for very different purposes, and the direct purpose of the class in question is the measurement of water. Therefore the term "steam hygrometer" would be more correct.

Separation of moisture from current of steam.—Steam separators are largely used for extracting the liquid water from a current of steam in motion. All these include an arrangement for suddenly changing the direction of the current of steam. The dry steam, being of low density as compared with that of the liquid water, more readily responds to the diverting force, while the liquid water continues moving along the original direction until it comes into contact with the casing of the receptacle, or with a plate or grating arranged to receive it, and to provide ready means of drainage. Obviously the precipitated water should be at once removed from contact with the current of steam, so as to avoid all possibility of re-union, and the drainage from the collecting plates should be effected without the exposure to the current of steam of any agitated surface of liquid or of falling drops. The collected water may be either returned to the boiler while still under pressure, or if the levels and conditions prevent this, it may be extracted by means of steam traps.

Collection of true sample of steam.—The collection of a sample of steam for testing as to the presence of moisture should be effected in such a manner as to avoid separation. The collecting pipe should run along the main steam pipe for a short distance, both being quite straight, and the orifice of the collecting pipe facing the current. The velocity in the main pipe and in the collecting pipe should be approximately equal. If the orifice of the collecting pipe is placed in an elbow and facing the current, the sample will probably

contain more than an average proportion of moisture. If the collecting pipe crosses the current of steam, and has holes distributed over it, some will collect steam too wet and others too dry, and possibly the average may be about correct. But this can only be assumed. In many cases the conditions necessary for absolute accuracy cannot be secured. But in all cases they should be kept in view, so that no serious error may arise.

Drying of steam by superheating.—Moisture in steam may be evaporated so that the whole assumes the form of dry steam. This may be effected by superheating the entire body of steam by passage through a superheater, by the addition of a suitable proportion of superheated steam to the current of wet steam, or by the outward application of heat, by means of steam at a pressure and temperature greater than those of the steam operated upon.

Temperature observations. — All processes whereby the proportion of moisture in steam is ascertained by means of observations upon the temperature of condensed water or throttled steam, will show also whether steam is superheated. This is, however, not the case in processes whereby the dryness is ascertained by the absence of separated water.

CHAPTER VII.

FORCED DRAFT.

Intensity of combustion increased by draft.—It is a matter of common knowledge that an improved draft causes increased combustion in a fire. Such improved draft may be secured by means of increased efficiency of chimney, or by the use of a mechanical blower, fan, or bellows. The absolute amount of heat developed per pound of fuel is not by such means increased, but combustion is completed in less space, more promptly, and before the gases can lose any important amount of the generated heat, either by radiation or conduction. The condition of rapid motion in which the gases exist when air is forcibly applied, is highly conducive to complete admixture, whereby less excess of air becomes necessary for securing complete combustion. The heat is consequently conveyed by means of a less quantity of gases at a higher temperature than under ordinary conditions, and the efficiency of the heating surfaces is increased.

Occasions for use of artificial draft. — About 20 pounds of coal per square foot of grate surface may be burnt by means of unassisted chimney draft, of strength indicated by a column of water of from five to seven-eighths of an inch. This varies with the coal, small

dirty coal requiring the best draft. When the amount of work required to be done is such as to call for a consumption of coal very much exceeding 20 pounds per square foot of grate surface, some artificial assistance must be given to the draft. Steam jets or blowers may be used for moderate assistance. These should always be applied to inject air into the ash-pit, where the steam is useful in preserving the grate bars from excessive heat, but the heat thus absorbed is restored at the upper part of the fire. A very large increase in the work may be obtained by the use of a blast-pipe as in a locomotive engine, supplied by the exhaust steam from a non-condensing engine. This measure is, however, attended by results too extravagant for general adoption upon stationary boilers.

Forced draft below and above fire.—Probably the most efficient means for increasing the work of a furnace consists in supplying air under pressure to the ash-pit and also above the fire, the pressure being obtained by mechanical means. Any kind of blower, such as Root's blower, would answer quite well. But in very few cases is the air-pressure required to exceed one-half to three-quarters of an inch column of water, and the quantity of air required to be supplied is always large. For these reasons a fan is best, and this should be provided of large diameter, so that the speed of revolution may be kept low, and the fan driven directly. The supply of air to the ash-pit should be so adjusted as to secure sufficiently powerful combustion in the lower part of the fire, but to avoid burning the grate. The supply above the fire should just suffice to effect the complete combustion of all gases which escape combustion within the fire. In Howden's system the air supplied above the grate is discharged in jets of small diameter, which are directed closely along the surface, so as to make

searching contact with the hot fuel. The jets are directed slightly downwards, but not sufficiently so to cause rebounding, by which they would become much less effective. With a similar object, deflectors may be adopted on the principle of Martin's furnace doors. Valves should be provided, and arranged so that the blast shall be shut off before the furnace door is opened, or the fire may possibly be blown into the stoke-hole, and cause injury to the attendants. When the air supply is thus cut off, the furnace is cleaned without any rush of cold air through the open door, which is a great advantage. The fire may also be brought into good condition after cleaning, in much less time and with greater ease and certainty than is possible under natural draft.

Possible use for prevention of smoke.—The higher temperatures and the more perfect combustion which naturally follow from the adoption of forced draft are of great advantage in reducing or preventing the production of smoke. When the principle is adopted with a view to the prevention of smoke,—as indeed in all cases—the grate area should be correspondingly reduced, either by shortening each grate, or by reducing the number of boilers under steam. Otherwise the forced draft is of no avail, and may even lead to increased smoke production by reason of overloading of fires which thus becomes possible, and is practised by unscrupulous firemen.

Possible use to meet fluctuations in work.—Forced draft is to be adopted with the greatest advantage in cases where the work is fairly uniform. When the work is irregular the defect mentioned in the last paragraph is apt to become developed during times of slack work. In careful use, however, it provides a most convenient and valuable means whereby fluctuations may

be promptly met. The supply of air may be automatically regulated by means of the arrangement sometimes adopted for the dampers of boilers under natural draft.

Use of forced draft allows more complete abstraction of heat from gases.—In a chimney working under natural draft, the temperature of the gases must be sufficient to give rise to the draft, which depends upon a light column of gas within the chimney. The whole of the heat thus accounted for is wasted so far as evaporative results are concerned, but under the conditions, such loss is quite unavoidable. Under forced draft, however, there is little or no necessity for the gases to escape at a high temperature, and the heat may be usefully abstracted more completely than is the case otherwise. In Howden's system, which has been chiefly though not exclusively applied to marine boilers, a blast heater is adopted. In this the escaping gases are led through a series of tubes, around which the fresh air is circulated, and acquires heat from the escaping gases. This interposes a double resistance in the circuit of gases, one element in the resistance to the escape of the spent gases and another in the resistance to the current of fresh air. By such means much more heat is abstracted from the gases than suffices to drive the blast engine. But stationary boilers are usually provided with economizers, and less heat is lost in the chimney than is the case in marine boilers. Consequently there is a less margin available, and less probability that the partial recovery of the remaining heat in the gases will prove commercially successful. Approximate calculations may, however, be made upon all conditions applying to a particular case, except as to the durability of the apparatus, which can only be surmised.

Forced blast prevents leakage of air into flues.—When

forced draft is adopted, the leakage of *cold* air into the flues through defective brickwork, damper slits, and economizer fittings is avoided. Any leakage which takes place will be in an opposite direction, and will be very likely to receive early attention. In a similar way, when an excessive pressure in proportion to the height or length and area of chimney and flues is adopted, and the heat extracted so far that a low temperature prevails in the chimney, it will be necessary to close the damper of each furnace before stoking, so that no return of the gases into the stoke-hole may arise.

Closed stoke-hole.—In marine practice the entire stoke-hole is often enclosed, and supplied with air under pressure. By this means the provision of separate fittings upon each furnace front is avoided, so that the cost in this respect is reduced largely. But this saving is much exceeded by the cost incurred in the air-tight construction of the stoke-hole. The chief object sought in the adoption of such practice is the reduction of weight of boiler and the saving of space in proportion to the power developed. The efficiency of combustion is almost equal to that obtained in the divided system, when rather more pressure is applied over the fire than in the ash-pit. The adoption of a closed stoke-hole is attended with much inconvenience and the production of much dust. But its weakest point consists in the rushes of cold air which occur when the furnace doors are opened. These are injurious at all times, but eminently so when this plan is adopted. This difficulty could be met in connection with stationary boilers by means of a system of dampers, but under the whole of the conditions it is scarcely possible to imagine a case in which it would be wise to adopt a closed stoke-hole in stationary work.

Draft by exhaustion.—In rare cases a fan is applied to draw the gases from the furnaces and discharge them up the chimney. This practice also is likely to lead to the admission of injurious quantities of cold air, and appears to possess no special advantage. The term forced draft, generally applied to the whole system of mechanical assistance or replacement of chimney draft, applies more strictly to this plan, and the term forced blast to all others.

Application of forced draft to existing boilers.—The adoption of forced blast to an existing range of boilers which are fairly up to their work under natural draft cannot be expected to largely affect the economical result. The same amount of water would be evaporated by nearly the same amount of fuel burnt upon a grate area one-third to one-fourth less by means of say three-quarters of an inch air pressure. Practically the same amount of heat would be produced, and the same amount of heating surface required for its economical absorption. By means of accurate adjustment of the air supply, a less amount of air may be supplied than is practicable under natural draft, the limit of economy to be thus secured being deducible from Table II., p. 37. As already explained, any reduction in the quantity of air supplied, while perfect combustion is secured, will cause increased temperature in the furnace, which again causes an increase in the efficiency of heating surfaces. There is also less heat carried off by all the gases through the chimney at a given temperature. No advantage is, however, to be secured in this respect, except by unremitting vigilance. By vigilant attention, the production of smoke may almost always be avoided. But the adoption of forced draft will be found to give greater facilities for the attainment of such results.

As in all other cases, the best results can only be secured when the furnace gases are regularly subjected to analysis.

Adoption of forced draft to new boilers.—The adoption of forced draft allows the use of smaller boilers than would be possible under natural draft. But a reduction in the size of a boiler naturally involves a reduction in the heating surface, which may be carried so far as to interfere materially with the efficiency of the boiler. A good economizer should be adopted for the extraction of the utmost possible amount of heat from the waste gases. A heater for the fresh air may thus become desirable in addition to the economizer; and in the absence of an economizer, it may be ultimately found desirable to adopt regenerators of chequered fire-brick, as in connection with metallurgical furnaces.

Application to multitubular boiler.—An increase in heating surface is more efficiently secured in a multitubular boiler—such as the ordinary marine boiler—than in a Lancashire or Cornish boiler. The tubes in such boilers may be rather smaller in diameter under forced draft than under natural draft; but the spaces between the tubes should be increased rather than encroached upon, as they are seldom excessive under natural draft, while under forced draft more steam is produced, and a better circulation of water is of the greatest importance. Multitubular boilers generally occupy much less space than Lancashire or Cornish boilers. Consequently when space is of importance, such boilers under forced draft are naturally likely to secure adoption on account of the double advantage.

Fan for producing draft.—The fan for producing the forced draft may be driven by means of a belt or rope from a separate engine. Usually the belt is better on

account of the high speed and the less weight in proportion to power transmitted; therefore the loss of power by reason of centrifugal force is less than it would be if a rope were adopted. But a small special engine coupled to the fan-shaft for direct driving is best, when the diameter of the fan is sufficiently large. It is less liable to interruption of work, it is more compact, and it may be started and stopped at any time. Such an engine is required to run at a high speed, and requires to be well balanced as to weights of moving parts and steam pressure so as to secure efficiency, durability, and absence of objectionable noise in working. Lubrication should be thoroughly reliable as to every point, and precautions adopted against scattering or spraying of oil. Every part should be promptly accessible in case of accident, and with a minimum of disturbance of details. Economy in the use of steam should receive some attention, though the quantity of steam consumed is not likely to be large.

Finish of pipes.—All pipes used for the conveyance of air should be made smooth and uniform inside, both as to area and surface. All pipe bends and valve passages should give an easy lead for the air in its passage. Valves should be free to work, and arranged to give full area when open. Leakages of air are to be avoided, chiefly on account of the liability to raise a cloud of dust.

CHAPTER VIII.

GAS FIRING.

Chief motive for application of gas firing to steam generation.—In connection with the generation of steam, gas firing is chiefly of interest on account of its providing the means for the utilization of classes of fuel which are quite impracticable for use in ordinary furnaces. Coal containing 50 per cent. of incombustible matter has thus been successfully used. Coal-dust, slack, lignite, peat, sawdust, shavings, and carbonaceous refuse generally are all suitable for utilization in this way. In some cases a stream of coal-dust is run into the gas as it leaves the fuel in the producer, at a sufficiently high temperature, and any remaining carbonic acid is thereby reduced to carbonic oxide.

Control and adjustment of supply.—The air and gas admitted are under the most complete control, and may be adjusted with the greatest accuracy, so that complete combustion may be secured with air in excess by about 10 per cent., while with hand firing, under ordinary conditions, air is required to be in excess by about 50 to 100 per cent. to ensure complete combustion. The fuel and the air are brought into such intimate contact, that even the excess of 10 per cent. could be reduced

except for the occurrence of slight variations in the composition or pressure of the gas, which arise from time to time. The perfect combustion which occurs leads to better economy, though the exact proportion of this varies largely in different cases: 10 or 20 per cent. has been secured, and in 1856 a saving of 30 per cent. was stated to have been secured at Cherbourg. The higher figures are naturally more likely to be obtained in comparison with plant which is not perfect of its kind.

Direct production of gas.—The gas used for firing is generated in producers, of which a very great variety has been brought into use. The two most important constituents are carbonic oxide and hydrogen. Carbonic acid is first formed by the action of the oxygen of the atmosphere upon incandescent carbon in the fuel. The carbonic acid then becomes reduced to carbonic oxide by subsequent contact with additional carbon, and this carbonic oxide, diluted with a certain amount of nitrogen from the air admitted, constitutes the primary "producer gas." Hydrogen, carburetted hydrogen, and carbonic acid are, however, always present in varying proportions.

Alternate production of gas.—Some producers are worked under an alternate system. In one period steam is admitted to the fuel in the producer. The hot fuel causes the steam to become decomposed, its hydrogen is set free, and its oxygen combines with the incandescent carbon to form carbonic acid, which is at once reduced to the form of carbonic oxide. An inflammable mixture of carbonic oxide and hydrogen is thus produced free from any important amount of nitrogen, carbonic acid, or other inert gas. The injection of the steam amongst the fuel, and the chemical action described, usually cause the absorption of heat, so that the temperature of the fuel is reduced. Before

this action proceeds too far, the supply of steam is cut off, and air is admitted to freshen the fire, in which process carbonic oxide is largely produced, after which the admission of steam is resumed. There are thus two processes alternately proceeding in the same producer, and two kinds of gas produced, which are known as "water gas" and "producer gas." These are, however, usually led into the same gas main for consumption, and to make the product more uniform two producers are employed, only one of which takes steam at the same time.

Presence of water gas from use of steam injection of air supply.—In some cases a steam jet is used for the injection of the air supply in making producer gas, so that some water gas is found in the product. The quantity of steam so admitted is, however, insufficient to interfere with the chief process of combustion. Being admitted at the bottom of the producer, such steam tends to prevent the fusion of the ash and facilitate its removal, while it would otherwise form clinker.

Use of water gas.—The water gas produced in the intermittent processes is claimed to possess superior heating power by reason of the low proportion of nitrogen which it contains. Indeed if the water gas is kept separate from the ordinary producer gas generated in the alternative process, it may be obtained practically free from nitrogen. The manifest advantages to be derived from a reduction in the amount of nitrogen present in the gases fail, however, in this case, for the reason that the full amount of nitrogen necessarily comes in, at the time of combustion in the furnace, at the moment when its absence or reduction would be beneficial. The order changes, but the precise amount of nitrogen is practically unaffected.

Downward collection of gas.—In the majority of

producers the gases rise through the fuel as generated. In the Loomis producer, however, the gases are collected beneath the fuel. This causes decomposition of carburetted hydrogens, which are therefore found only in small quantities in the product.

Division of combustion into two stages by use of producer.—By the use of fuel in a gas-producer, its full and complete combustion is effected in stages. The first part is effected in the producer, in which some heat is generated, of which part is absorbed or stored in secondary chemical processes, to be afterwards restored to the general stock. Some is also expended in the physical gasefication of the several substances produced. The largest amount of heat is, however, generated in the furnace into which the gas is subsequently introduced simultaneously with the proper amount of air to effect combustion. The total obtainable amount of heat remains absolutely the same, whether the process is completed in one stage or divided. The use of steam cannot largely affect the question, for the reason that though the combustion of hydrogen, or its combination with oxygen in the furnace to form water, generates a large amount of heat, yet the separation of the same elements, which is previously effected in the producer, absorbs precisely the same amount of heat. Therefore the total amount of heat produced in the combustion of a certain amount of fuel cannot be increased by the use of a producer, or by the assistance of steam. Any other condition would involve the creation of energy, which is only one point removed from the production of perpetual motion. The production of water gas should also be debited with the fuel consumed in raising the steam before its supply to the gas-producer. In precise calculations also, the heat carried off by the produced water will be found to be important.

Intermediate loss of heat and use of cooling tube.—It is evident that if producer gas be allowed to cool between the several stages, the amount of heat rendered available in the final process of combustion will assuredly be less than it would have been under the direct process of combustion of the solid fuel. Some such loss arises from radiation and conduction from the surfaces of the producer and gas mains. Such loss was formerly also intentionally incurred in the adoption of a cooling tube for the purpose of propelling the gas along the mains. This consists of a horizontal tube placed at a high level, connected at one end to an upcast tower leading from the producer, and at the other end to a downcast tower leading to the furnace. The horizontal tube is arranged to facilitate the cooling of the gases, so that the density of the column in the downcast tube shall exceed that of the column in the upcast tube, sufficiently to cause the required current in the gas. In some cases this cooling has been carried so far as to effect the condensation of moisture contained in the gas.

Steam blast.—As a substitute for the cooling tube and as a means of assistance to the producer, blast is in present practice usually supplied by means of a steam jet injector. By this means a large loss of heat is avoided, and the condition becomes very similar to those applying to the adoption of forced draft for other purposes. Greater freedom is also experienced in the location of the producer, which is otherwise required to be placed at a low level.

Wear and tear.—Gas-producers of all kinds are subjected to great wear and tear. As in other furnaces the brickwork is affected in three ways. (1) The mechanical abrasion caused during stoking and breaking down the fuel and in removing clinker, acts very

destructively upon bricks softened by the application of heat. Some action of the same kind is also due to hard fuel or stones, &c. accidentally received. In some producers the fuel exists in the form of a tall column, the weight of which suffices to effect the feeding and trimming of the burning mass, when the use of stoking tools is quite exceptional. (2) The ash contained in the fuel often possesses the properties of a flux, as affecting the bricks. The two substances possess a chemical affinity for each other at furnace temperatures, and in their combination produce a substance which is fusible at the temperature to which it is exposed; it therefore melts and becomes lost in the clinker. It is practically no less serious when a similar relation exists between the ash and the cementing material used for bedding the bricks. Naturally the bricks exposed to contact with the incandescent fuel suffer the most severely from this cause, but much damage of the same kind is often caused to other parts by reason of fine particles of ash carried forward by the draft, and becoming attached to any heated surface, especially those exposed to the direct impact of the hot gas. (3) Bricks often suffer by alternate exposure to different temperatures, especially when the changes in temperature are rapidly effected. In this respect substances of the most homogeneous constitution are found to be the most successful. When the minute structure of a brick is composed of crystals or particles of diverse nature, the expansion of adjoining particles when exposed to heat almost always varies; and fracture inevitably follows sudden and extensive changes in temperature. Gas-fired furnaces are exempt from the first and second means of destruction, but are exposed to the third. In all cases the bricks or other lining

adopted must obviously be quite infusible at the temperatures to which they are exposed.

Stoking producers and removal of ashes.—Means should be provided whereby ashes may be removed, or fuel added, or stoking effected, above or below the fire, and breaking down the fire performed without any interference with the supply of gas from the producer, or the occurrence of outbursts of gas or explosive mixtures of gas and air. The importance and difficulty attaching to these conditions vary with the class of fuel and the length of run, but on the whole they are best met by the adoption of natural draft, and sufficiently tight construction to avoid inward leakages of air. Uniformity of gas is also a very important point, only to be secured by strict supervision, and uniformity in feeding the producer, and in the preservation of a solid fire, without holes for the passage of air. Cleaning out is best effected during stoppages for meal hours or after working hours.

Production of tar.—The volatile hydrocarbons evolved by the green fuel sometimes condense in the form of tarry matter in the mains and about the valves. In some producers, a curtain or hanging tube of brickwork is arranged, to cause the hydrocarbon vapours to pass through the incandescent fuel below on their way from the producer, and thus become decomposed. This is effectually done, but the curtains are troublesome in maintenance. In the Loomis producer, as already stated, this is effected by reason of the collection of the gas below the fire. The tarry matter is the objectionable element in the gas from many producers, which affects its value for use in gas engines, on which account gas for this purpose is generally produced from coke or anthracite.

Combustion of gas.—Gas is admitted to the furnace in which it is consumed by a series of ports, and the air which is necessary for combustion is admitted by a second series. These must be so placed and directed as to cause an intimate admixture of the whole; the ports should also be symmetrically disposed, so as to avoid deflection of the flame, which interferes with perfect combustion. Gaseous fuel gives great facility for the concentration of great heat at one place, which facility is of great importance in metallurgic processes. With this object, sufficient space should be provided so that the process of combustion shall be completed, and the heat developed before the gases leave the furnace. In boiler firing, however, the necessity for this does not exist; the gas and air should therefore be brought into contact more gradually, so as to reduce the tendency to overheating and destruction of the surfaces in contact with the flame. In any case, however, the completion of combustion must not be delayed so long as to cause loss of unburnt gas by reason of cooling. The flame produced by water gas is small, and yields an intensely concentrated heat, as compared with a more diffused heat obtained from carbonaceous producer gas. Consequently the latter is much to be preferred for boiler work.

Pipes and fittings.—In gas work the pipes and fittings should be securely bedded and jointed, and every precaution taken throughout to prevent leakage. Leakage is wasteful, and apt to lead to damage by explosion, and with inodorous gas may lead to loss of life. Troubles of this kind are much more likely to follow if the gas is worked by forced draft, and may be almost, or entirely, absent if the gas is under a slight vacuum. In all cases, however, good fitting of all details should be insisted upon.

Disposal of ash.—The facility presented by the gas

firing principle for the use of inferior fuel often leads to the production of large quantities of ash. This fact in connection with the large quantities of fuel required should be considered in deciding upon the position of producers.

Gas firing for heavy production of steam.—In the adoption of gas firing for heavy powers, a good draft will be found to be necessary to draw the gas and air through the supply ports, otherwise difficulty is likely to be experienced in obtaining the required power. Similar difficulty is also likely to arise by reason of vitiation of draft power by reason of entry of air in irregular ways. Therefore in case of any little difficulty the first steps should be to investigate the sufficiency of area of ports, according to the principles applied to chimneys, and to subject the flue gases to analysis.

Gas firing for metallurgical work.—Hitherto this principle has been most largely adopted in connection with steel making and other metallurgic and allied processes, and also incidentally for purposes of steam production in connection with these. For the former work, temperature is usually of greater importance than absolute volume of heat, and the greatest advantage of gas firing consists in the facility with which very high temperatures may be maintained when adopted in combination with regenerators. These usually consist of chambers built of fire-brick, and filled with fire-bricks built with chequers or spaces, through which the waste gases are passed, with a view to impart their waste heat to the brickwork, after which the air supply, on its way to the furnace, is passed through the heated brickwork in the opposite direction, and takes up the heat stored in the first operation. Two regenerators are used and the actions alternated in each, so that hot air is continuously supplied to the furnace, and corre-

spondingly greater temperatures secured than would otherwise be practicable.

Conditions applicable to boiler firing.—This system is, however, less adapted for use in connection with steam boilers on account of the lower temperatures existing in the boiler flues. The action of the ordinary economizer, or feed heater, is continuous, and requires less attention than regenerators which require to be reversed at short intervals. If, however, any attempt should be made to extract the waste heat from the gases more completely than is at present done, there is some probability that brickwork regenerators will be found useful, not to replace, but to supplement economizers.

General advantages.—Though the advantages attendant upon the adoption of gas firing of boilers are few, in some cases they are very powerful. They consist in the chemical efficiency of the process, in allowing a narrow margin of air supply, and the use of inferior fuel. Gaseous fuel may be supplied in exact proportion to the work. It is brought into the most intimate possible contact with the air, free from the obstructions of grate bars, solid pieces of fuel, clinkers, &c., and the process of combustion is effected promptly and efficiently. The fuel may be brought to the point of application in pipes without interference with traffic above ground, and no ashes require removal from the same point. Gas firing also allows no excuse for the production of smoke.

Producer gas for gas engines.—Many gas-producers on a comparatively small scale are used in connection with gas engines. This manner of producing motive power from coal evades certain losses of heat, which are elsewhere referred to. The experience obtained with gas-producers for this purpose may ultimately lead to such improvements as to cause an important increase in their use in connection with steam production.

References.—Reference may be made to a paper on cheap gas for motive power by Mr. Dowson, in the *Minutes of Proceedings of Institute of Civil Engineers*, vol. lxxxiii. p. 311; also one on gas-producers, by Mr. Rowan, in vol. lxxxiv. p. 2; also to *Engineering*, vol. xxix. p. 2, in which a number of furnaces are discussed.

CHAPTER IX.

USE OF LIQUID FUEL.

Descriptions and qualities of liquid fuel.—Liquid fuel for steam generation chiefly comprises crude petroleum, petroleum residue,—which remains after the removal of the more volatile constituents from crude petroleum,—crude shale oil, shale oil residue, and tar obtained in the manufacture of gas from coal. Many other substances are used on a smaller scale, or in special cases, but practically the whole are treated alike. The constitution and composition of these vary largely, but in all cases the proportion of hydrogen is considerably greater than in coal, and naturally the calorific value is greater than in fuels which contain less hydrogen. The combustion of carbon in a liquid state renders available a larger amount of heat than does the combustion of an equal quantity in a solid state. This is because more heat is absorbed or rendered latent in the conversion of carbon from a solid state to a state of vapour—however this may be effected—than in its conversion from a liquid state into vapour. Consequently in the use of liquid fuel, a larger proportion of the heat generated in the consumption of a given quantity of carbon and hydrogen is available for useful application than is the case in the use of solid fuel. In the use of natural gas

a still greater advantage is similarly secured. But in the use of gas generated from solid fuel, such advantage is only secured at the cost of heat directly expended in the process of generation.

Efficiency of combustion.—Liquid fuel is supplied to the furnace in a finely-divided condition, in intimate contact with air, by which means it is possible to ensure complete combustion without the presence of such a large excess of air as is necessary in the use of coal or other solid fuel. For the several reasons given, it is found that the combustion of one pound of liquid fuel will produce from one and a half to three times the amount of useful heating effect which is produced in the combustion of one pound of coal.

Injection of supply to furnace by means of steam.—Liquid fuel is chiefly utilized in practice by injecting into the furnace, by means of a current of high pressure steam, preferably superheated. This operation is generally effected by means of an instrument which in the main principle is equivalent to an ordinary boiler injector. A jet of steam is directed in contact with a supply or jet of fuel, in such a manner that the latter is carried into the furnace by the steam in a finely-divided or "pulverized" condition. In different systems the jets are differently arranged, but in most cases a central —solid or annular—jet of steam is surrounded by an annular jet of oil. This usually works with less risk of stoppage than other types. In other cases, delivery is effected in a long fine line, either from an annular opening or along a straight line, with a view to an increase in the fineness of division. In all cases, a free supply of air is furnished, so disposed as to be drawn into the furnace by induction, and become thoroughly blended with the pulverized oil, by which efficient combustion with a minimum quantity of air is ensured. In some cases this

operation is promoted by the supply of air through a third channel or series of apertures in the pulverizers, but in most cases the air is simply allowed to enter around the whole instrument. The instrument should be so constructed as to prevent the accidental mixture of steam and oil before reaching the intended point of combination. The pulverizer should not be unnecessarily exposed to heat, which reduces the life of the instrument, and increases its liability to stoppage.

Supply by distillation.—In some other systems the liquid fuel is subjected, in a special vessel, to distillation, after which it is supplied to the furnace as gas; or the less volatile parts are carried along in the form of spray by the more volatile, which are in the form of vapour. Steam may also be used to assist the supply. The appliances necessary for the adoption of this plan are very costly, some considerable amount of danger is attached to their use, and no material advantage is secured.

Supply by compressed air.—Liquid fuel is also supplied to furnaces by means of a jet of compressed air. This gives very good results, but is subject to occasional stoppage, probably on account of the actual process of combustion occurring nearer to the nozzle, so that the latter is subjected to greater heat than is the case when steam is used. In this way any impurity which may lodge in the aperture of the nozzle becomes charred, and remains in the aperture. In the absence of steam, compressed air furnishes a convenient means whereby the supply of oil to a burner is maintained. In this way it has been largely used for some of the burners, in which crude oils are burnt to produce a strong light. It has been largely adopted in the United States for metal furnaces, but not extensively in connection with steam production.

Furnace temperatures.—The high calorific power of liquid fuel, and the moderate amount of air with which it may be supplied, cause the production of a furnace temperature very much higher than is possible with the use of coal, and in the absence of appliances for the regenerative treatment of the waste gases. The flame which extends from the pulverizer is of such a form that if directed upon a heating surface for steam production, the intense heat may be applied over a comparatively small surface. This application is therefore both severe and irregular, and such surfaces are found to suffer to a very serious extent. Hitherto liquid fuel has probably been applied most extensively to boilers of the locomotive type, in which, in the absence of brickwork, it is almost impossible that large pointed flames can be applied without impinging upon a surface. In some of the most successful examples, a chamber or oven of fire-brick is constructed inside the fire-box. The flame or flames are then directed into this chamber, and reverberate back, by which means the gases attain a more uniform temperature, and perfect combustion is assured. In others, the same object is secured by the provision of a fire-brick arch. The heating surfaces are still exposed to a very high temperature, but its uniformity and the purity of the flame cause the metal to suffer less than from gases of a lower temperature produced by the combustion of coal. In connection with boilers of other types, there is less imperative necessity for the adoption of ovens of this kind, but still they are of great value. Chequered fire-brick walls answer very well, especially if made solid in way of the flame of each burner, so that the intense local heat is not allowed to pass through and impinge upon any plate beyond. Fire-brick arches and walls are also of service in protecting the flames from cooling before

the completion of combustion and causing rekindling after a short stoppage, as when, in a locomotive, no steam is required for a few minutes. Sometimes a thin slow fire is kept alight so as to prevent the sudden extinction of the flame, but this is seldom really necessary.

Precautions to be adopted in kindling.—In kindling the liquid fuel, a small flame should be lighted in front of each burner, then the steam admitted, and afterwards the oil, when the whole will ignite. The oil and steam are then adjusted so that a clear, smokeless flame is produced, though this may probably not be completely secured before the brickwork is heated up. If the oil is admitted before the steam, or if the two together are admitted for a short time before they are ignited, there is some risk of an explosion, which may be of great violence or only a slight puff.

Supply by trickling stream.—In exceptional cases tar or other liquid fuel may be fed into a furnace in a stream, from a pipe or spout, and allowed to drop from a height upon a bed of breeze. The vertical distance dropped should be as great as possible, so that the individual drops are as much exposed to the air as possible. In this system there is more liability to intense local heating than occurs with the use of a pulverizer. Fire-brick suffers very much, and the heating surfaces of boilers would be found to give great trouble. There is also some considerable difficulty encountered in the prevention of smoke.

Precautions in storage.—Crude oil contains freely volatile constituents which are not contained in residues, having been removed by distillation. Both are of practically equal efficiency when properly used, but the former is more liable to accident by fire or explosion. These may, however, be avoided by the adoption of reasonable care in the construction of the fittings and

in the operation of the same. Most of these fuels are so heavy as to sink in water, for which reason they are much less dangerous than others which float upon water, and which are consequently more difficult of extinction when an accidental fire occurs. The main supply tank should be constructed below ground. A small tank should be used into which the current supply is pumped. The contents of the latter should be returned to the lower tank for greater security when work is finished for the day. Each pipe should be commanded by a valve conveniently situated for ordinary use, and also a second valve so connected as to be easily closed or tripped from a position outside the building. Both these valves should be closed regularly, so that they can be depended upon when required. All tanks should be well protected from the access of dust. Dust of all kinds is exceedingly difficult to separate by subsidence from liquids of the classes under consideration. If it fails to become separated, it is almost certain to give trouble by lodging in the apertures of the pulverizers. If it should become separated, it collects in a bed of mud in the tank, which reduces the practical capacity for storage, and which is difficult to deal with. In all cases fine strainers should be used to separate palpable impurities.

Advantages of liquid fuel.—If liquid fuel were equally abundant with coal, and were not more costly in proportion to the quantity of heat obtained in its combustion, it would effectually displace coal for all purposes of steam generation. At the present time the supply is very far short of that which would be required for its exclusive use, but it is increasing slowly. Under present conditions, its economical use must depend chiefly upon the relative prices of the two classes of fuel in any particular case. The Baku district at the

south-eastern corner of European Russia produces more liquid fuel than any other district in the world. At the works, the petroleum residue is worth about ten shillings per ton, at which price comparison with coal is out of the question. Its abundance in this case is due to the large production of crude petroleum, and to the large proportion of residue which is obtained from it, in comparison with American and other crude petroleums. In districts situated at a distance from petroleum fields, some substances are produced which may be used as liquid fuel, such as shale oil residues, tar from gas works, &c. The use of these, as of petroleum residues, depends largely upon the market value on the spot and at the particular time. But other elements enter into the consideration of the question. The supply of liquid fuel being by pipes, causes much less interference with other operations than is caused by the transit of coal. This reason had a considerable influence in the selection of liquid fuel for steam generation at the Chicago Exhibition. Liquid fuel may also be stored in any position, or at any distance from the point of consumption, without the necessity for any permanent means of access or communication other than a pipe. In this way a very great saving in space in front of the boilers is secured. A ton of liquid fuel occupies about 36 cubic feet; a ton of coal is assumed to occupy 40 cubic feet, but may be closely packed in rather less, so that the two may be regarded as equal. Liquid fuel for the production of a given amount of heat may therefore be stored in a much smaller space than that necessary for an equivalent amount of coal. No ashes are produced from liquid fuel, so that savings are effected in comparison with coal firing by reason of labour involved in their removal and in space for storage.

Fig. 1 shows, in longitudinal section and end elevation, a burner designed by Mr. Holden of the Great Eastern Railway, in which the jet of oil in contact with the supply of steam is kept very compact, whereby the liability to stoppage is minimized. The stream is dispersed through a triple nozzle, and is met by jets of steam directed upon it from a belt as shown. By this means also a supply of atmospheric air is ensured for combustion, and atomizing and distributing the liquid fuel entirely over the furnace, whereby intense local heating is avoided, and also the necessity for arches

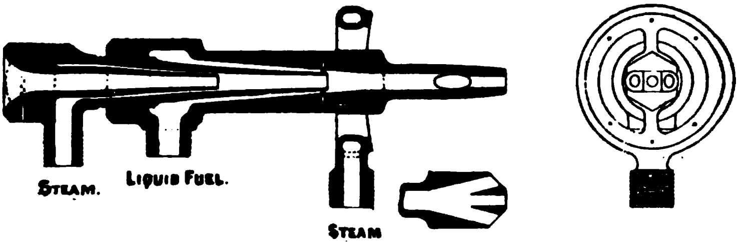

Fig. 1.—Holden's patent liquid fuel injector, stationary pattern.

and diaphragms of brickwork. Mr. Holden uses liquid fuel and coal in the same furnace, either alternately or combined in various proportions. A furnace is arranged with grate bars of the ordinary pattern, upon which a layer of chalk is spread, and upon which the coal to be burnt in combination with oil is dealt with in the ordinary manner. The fire-brick bridge is carried rather higher than in usual practice for coal burning, and is "pigeon-holed" or perforated. The furnace door is divided into two halves meeting in the centre, with a circular opening cut out, in the centre of which the fuel injector is placed, with sufficient space around it to admit the required air. An overhead tank is pro-

vided to contain the current supply of liquid fuel, from which it is led to the injector by a cock and pipe. A steam supply mounting is attached to the boiler front, and provided with three cocks, of which two are connected by piping with the two steam branches of the fuel injector, and the third is connected with a warming pipe led through the tank for use in cold weather. The warming pipe is continued through the bottom of the tank to drain.

The loss of water by reason of steam supplied to pulverizers is in some cases said not to exceed one per cent. of that evaporated in the boiler, and is seldom a factor of importance. But in many cases more steam is used in this way than is necessary.

CHAPTER X.

ANALYSIS OF GASES PRODUCED IN COMBUSTION.

Objects to be secured.—Analysis of the gases produced in the process of combustion is undertaken because it furnishes the best possible evidence as to the efficiency with which the process is conducted. Ideal efficiency requires the exact and complete combination of oxygen with the combustible elements of the fuel, and that this shall be effected without any excess of combustible on the one hand or of oxygen on the other hand. The information obtained is useful at all times, but especially so when changes in plant are contemplated, or are in process, and is *necessary* when new plant is required to be worked to best advantage.

Excess of air.—The analysis may show that air is deficient, and that combustion is incompletely performed. In many cases this happens to such an extent that carbonic oxide is found in the waste gases. In very rare cases, hydrogen escapes in the gases, but it occurs so seldom and in such small quantities that it is generally considered to be undesirable to complicate the operation by processes for its determination. In the large majority of cases the air will be found to be in excess.

Volumetric analysis.—In researches of the highest

degree of exactitude the gases are analyzed by weight. But this course renders necessary the use of a chemical balance, and other apparatus of great delicacy, and the operations occupy a much longer time. For these reasons, practically all chemists adopt a method of analysis by volume. As a rule, the results of analysis are more useful when stated volumetrically, but for certain purposes they require to be translated to show the proportions by weight.

Components to be determined.—In the practical analysis of these gases, four components are required to be measured, viz. carbonic acid, oxygen, carbonic oxide, and nitrogen. A definite quantity of the mixed gases is measured, and the first three substances are removed in the order given; the respective quantities are ascertained from the measurement of the total volume before and after the removal of each. The quantity left after the three first-named have been removed is assumed to consist entirely of nitrogen. When hydrogen occurs in the gases it remains in the nitrogen, but for the reasons before stated, it may be neglected. Carbonic acid is produced in the process of complete combustion of carbonaceous fuels. Hence a high proportion is produced in good combustion. Carbonic oxide is produced in imperfect combustion, and hence it should be absent, or only present in small quantities. The proportion of oxygen indicates the amount by which the air supply exceeds the chemical requirements of the fuel. The residue consists of nitrogen which remains from the air, and is by far the largest constituent of the whole. The water which originally existed in the fuel, or which arises from the combustion of the hydrogen in the fuel, is condensed in the apparatus, without entering into the calculations.

Collection of gas.—A sample of gas may be collected by means of wash-bottles or aspirators of any kind. But care must be taken to ensure that the sample consists only of gas drawn from the flue. The sample should be drawn from about the centre of the clear opening of the flue. It is best done through a small opening purposely made, and in Lancashire and Cornish boilers, so disposed as to draw the gas from beneath the boiler at 10 or 12 feet from the front end. In this arrangement, any dilution of the gas with air, after leaving the furnace and before analysis, is avoided, but it is just as essential for economical reasons that subsequent entry of air to the flues should be completely avoided. As a check upon this condition, a second sample should be drawn from some point near to the bottom of the chimney. In an emergency, a sample of gas may be collected through a narrow damper slit, but the tube employed should be bent inside the flue, so as to face the current. In all cases the orifice at the end of the tube must be quite beyond the reach of any influx of air, through the opening through which the pipe is inserted. In exact boiler trials, a continuous sample is very slowly withdrawn during several hours, and a smaller sample drawn from the large one, either hourly or at the conclusion of the test. The whole of the connecting-pipes must be entirely cleared of air before the collection of the sample is commenced. Whenever possible, glass tubes should be used for the collection of gases. In some cases the temperature is too high for this, and in others they are required in lengths too great for glass. Glass tubes should be of specially small bore, with a view to reduce the volume to be cleared out, to give a steady flow and provide increased strength of tube. Iron tubes possess the advantages of strength, freedom from breakage, melt-

ing or softening at high temperatures, and portability. They are, however, subject to oxidation, whereby the percentage of oxygen in the collected sample is reduced, and those of the other constituents correspondingly increased. This disadvantage can, however, only become appreciable when the temperature is very high, the gases passed through with extreme slowness, and with oxygen present in large proportions. If the pipes are coated internally by the Bowen-Barff or other equivalent process, further oxidation becomes impossible. The possibility of oxidation would not be avoided by the use of copper pipes. If desired, pipes may be permanently connected, so that the furnace gases may be easily sampled at any time. If this is done, the most scrupulous care should be taken with a view to avoid such projecting pipe ends as would prove to be dangerous to a man passing along the flues in comparative or total darkness; and special care would be necessary—on account of the probable great length of pipe—to ensure complete clearing of the pipe before taking a sample.

In exact boiler trials, elaborate appliances are necessary for collecting a very large sample, to be again sampled; but for ordinary purposes two wash-bottles or flasks will suffice. These should be fitted with india-rubber corks to prevent leakage of air inwards, or of gas outwards, and should be filled with water preparatory to use. The bottle *a* to be placed in an erect position, with the long tube *b* withdrawn, so as to be just level with the inside of the cork. This tube to be connected with the pipe leading from the flue. The second bottle *d* to be inverted, and connected with the first as in Fig. 2, and allowed to empty by the pipe *f*. By this means a quantity of mixed air from the pipes and gas from the flues will

be withdrawn, equal to the capacity of the bottle *d*, when the connections *g* and *h* are closed by screw pinch-cocks. A retort stand with open-ended ring is useful for supporting the inverted bottle. In case the current should cease prematurely, it may be restored by lowering bottle *d*, so that the flexible pipe *h* should be made of ample length for this purpose. If

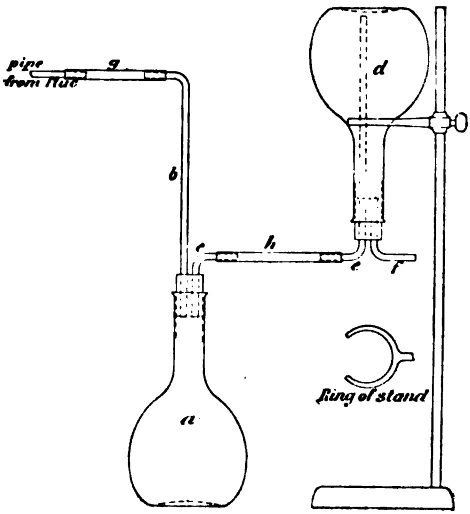

Fig. 2.—Glass flasks arranged for collection of samples of gas.

it is considered necessary to continue the clearing of the pipes, the bottle *d* is refilled and the operation repeated. When the pipes are deemed to be sufficiently cleared, the tubes *g* and *h* are again pinched, the tube *b* is pushed into its ordinary position, bottle *a* is inverted, bottle *d* is erected, the pinch-cocks removed from *g* and *h*, and bottle *a* filled with gas. By continuing the process, bottle *d* may also be filled

with gas. The pinch-cocks are then again closed upon the pipes *g* and *h*, and the ends of these stopped by solid glass stoppers, so as to seal the bottle *a*, the contents of which are then ready for use. The whole of the connections should be tested from time to time, to make sure as to tightness. One cause for leakage is the hard condition which india-rubber corks and tubes sometimes assume in cold weather. This is removed by slowly warming them and kneading in the hands or stretching. In the operation of collecting gas it will be noticed that small bubbles of condensed water will pass along pipe *b*. This water comes from the flue gases, and cannot be avoided, but it should be clearly distinguished from inward leakage of air around the tubes or corks, which must be stopped at once.

Fluid used in apparatus.—The use of pure water in the vessels employed for collection of samples causes some loss of gas by solution, which affects the composition of the whole, on account of the different degrees of solubility of the different constituents. But when the samples are required for immediate treatment, this action is much too trifling for serious consideration. If samples are required to be stored for any length of time, it is better to use a strong solution of salt for the purpose. Mercury is, however, best, except for the reasons that it is very costly, and its great weight is apt to cause accidents, whereby the whole may be lost and work stopped. When it is used an iron dish should be placed beneath any glass vessel containing it.

Description of burette.—A "burette" is used to contain the gas during the operation of analysis. This is graduated so as to allow the measurement of the volume at successive stages of the operation. Bunte's

improved form, as shown in Fig. 3, is probably used more than any other, but all are constructed in the form of a tube for facility and accuracy in estimating quantities. Others vary only in matters of detail. Bunte's instrument contains 100 cubic centimetres in the space between the upper cock and zero, distinguished by a cipher. The space above zero is graduated by divisions which are closer at the lower part than above. These divisions are carried below zero, so that it is possible to measure any quantity up to 110 cubic centimetres in the instrument. It is, however, most convenient to adopt 100 parts, so as to avoid calculation in ascertaining percentages. Chemists read the burette to the second decimal place in percentages. In ordinary cases it is, however, quite sufficient to read to the first decimal place—that is, to the one-thousandth part of the whole, and to half a division, as marked upon the stem of the instrument. The surface of the liquid in the burette will be seen to assume a meniscus or cup form. The reading should be taken at the bottom, and not at the edge, and in all cases a constant time of, say two minutes, must be allowed for the liquid to drain from the sides of the burette.

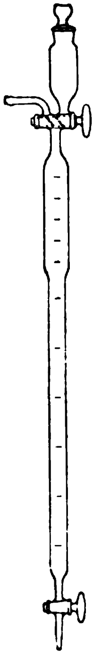

Fig. 3.—Bunte's burette.

Use of burette.—The burette is filled with water in preparation to receive the gas. An aspirating bottle is used for filling the burette, for convenience, and because the water in the aspirator acquires the temperature of the room, which is conducive to accurate working. The aspirator is fitted with a flexible tube and pinch-cock. In preparation for use, it is filled

with water and placed on a shelf, above the level of the top of the burette. The burette is supported vertically by a stand, in such a manner as to be readily removed when required. Assuming the burette to be filled with air, the flexible tube from the aspirator is pushed on the lower nozzle of the burette, both burette cocks opened, and the pinch-cock relieved. This will allow the water to flow from the aspirator into the burette, driving the air before it through the upper cock. The latter is a three-way cock, which either closes communication, or alternatively connects the body of the burette with the cup or the upper nozzle. In filling the burette with water, it should be charged to the point of the upper nozzle, and the cup should be filled to a definite level mark upon it. The burette is then ready to receive the sample of gas for analysis, which is brought in the bottle, sealed with a pinch-cock and plug in each tube. The flexible tube *h* is connected to the upper nozzle of the burette, and the flexible tube *g* connected with the tube of the aspirator, the end of each tube being filled with water to prevent any admixture of air—an object which should always be kept in mind. The pinch-cock should then be relieved, the upper cock of the burette opened to the gas, the lower cock opened for the exit of the water and the gas admitted under control of the lower cock. The slight pressure in the sample bottle prevents any inward leakage of air through misadventure in making the transfer. When the water level in the burette has run down to a little below zero both cocks are closed. The aspirator tube is connected to the lower cock and water admitted, until the gas is compressed to 100 volumes, the excess of gas being allowed to escape by the upper cock through the cup. The upper cock should never be opened to the cup when the

stopper of the cup is in position, as the withdrawal of the stopper is then liable to cause the escape of gas from the burette. The surface of the water in the cup should also be kept strictly to the level mark, so that by opening the cock to the cup, before reading off the volume in the burette, the gas within the burette may be always placed under a constant pressure equal to that of the atmosphere *plus* that due to the head of water in the cup. If necessary, a little more gas may be admitted, previously to commencing the analysis, but after this the greatest care should be taken to prevent either the entry or the escape of gas.

Exhaustion of burette.—After the volume of gas has been accurately verified under the pressure due to the head of water in the cup, the cocks are closed, and the volume read off and recorded. Some of the water contained in the burette is then removed through the lower cock by means of the suction bottle, and a partial vacuum created. The suction bottle (Fig. 4) is used by connecting the long tube of the bottle with the lower nozzle of the burette, by means of a piece of flexible tube. The suction tube of the bottle is then exhausted, and the lower cock of the burette gently opened. This must be closed again before the exhaustion is relieved, or when the liquid in the burette ceases to run down, otherwise fluid and possibly air from the tube will be returned into the burette and affect the result.

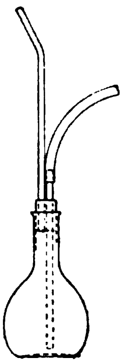

Fig. 4.—Suction bottle.

Removal of carbonic acid.—When the burette is in an exhausted condition, a quantity of solution of caustic potash is placed in a small dish, and held against the lower nozzle of the burette, while the cock is gently

opened and the solution drawn in. The burette is next removed from the stand, and carefully but vigorously shaken for two or three minutes, during which the carbonic acid is absorbed by the potash solution, and the total quantity of gas in the burette is reduced. The dish containing potash is again brought to the lower nozzle, and solution again admitted sufficiently to destroy the vacuum. The shaking is performed in such a manner as to secure the greatest degree of intermixture between the gas and the re-agent used, which is easily acquired by practice. The operation is repeated until no more solution is drawn in at the lower nozzle, the liquid is washed out, water is admitted from the upper cup, which is kept filled up to the level mark, the water in the tube allowed to drain down during the proper length of time, and the volume read off. The difference between this and the original volume shows the amount of carbonic acid absorbed. As much solution as possible should be drawn in at the lower nozzle before admitting water from the cup, as by this means the maximum strength of solution in the burette is secured.

Washing.—When the reaction is quite completed, the solution is removed by the suction bottle, and a stream of water admitted from the cup to wash out the tube as above referred to. To ensure that no loss of gas shall occur, the lower nozzle may be immersed in water before the lower cock is opened. Then attention may be concentrated upon the upper cup, to see that it is kept well supplied with water from the aspirator. By this means the film of re-agent is washed from the sides of the burette and an accurate reading becomes possible. If the water in the cup is allowed to run out, the seal will be broken, and gas may escape or air enter, in either case destroying the

value of the analysis. At the close of the washing process, the gas in the burette must be left under the pressure due to the head of water in the cup when just up to the level mark. Then time must be allowed for the water to drain from the sides before the reading is taken.

Precautions in connection with the entry of re-agent.—The admission of solution by the lower nozzle must be effected with care, so as to avoid the entrance of air. This is likely to arise if the nozzle fails to reach one-eighth of an inch below the surface of the liquid. When the dish is withdrawn from the nozzle, a drop of liquid remains attached to the latter. This should be removed by a cloth, otherwise the drop will probably fall away in shaking the instrument, and leave a small bubble of air to be admitted on next opening the cock. By this means also soiling of the hands is prevented.

Partial analysis.—In analyses undertaken only with the object of locating leakages of air into the flues, the proportions of carbonic acid found in the flue gases from different points will give the means required. In this way time will be saved, and a larger number of analyses may be made than could be possible if each were carried to the full extent.

Removal of oxygen.—After the removal of the carbonic acid in an ordinary analysis, the oxygen is removed by means of about three-quarters of a cubic inch of pyrogallic acid, added to the potash solution, which was removed from the burette by means of the suction bottle, and is now returned to the burette by means of the dish. The shaking is performed exactly as before, except that it is more prolonged; the solution is also added as before. The solution rapidly becomes dark-brown from absorption of oxygen, just as in

photographic work a pyrogallic developer turns dark. When the absorption has ceased, the removal of the solution, the washing, and the reading of the volume are effected precisely as before, the amount of oxygen being given by the difference between this reading and the last.

Removal of carbonic oxide.—Carbonic oxide is removed by means of a solution of cuprous chloride in strong hydrochloric acid. As this is precipitated upon contact with water, it is especially necessary that as much as possible of the water used for the washing of the gas shall be removed before the admission of the re-agent. Owing to the reduced volume of the gas at this stage, the removal by suction is more difficult than in the preceding stage. A quantity of water is placed in the suction bottle, and the tube filled with water up to the nozzle of the burette by blowing. The bottle must be placed on the base of the stand so as to leave the hands free, and the suction then carefully applied. If one pull is insufficient to empty the burette, the cock must be closed each time before the suction is relaxed. After the removal of the washing water, the solution of cuprous chloride is admitted by means of the dish. The burette is shaken as before, the test for completeness being made by the application of the same solution in the dish, and avoiding all contact with water until the final measurement of volume, which as before must be preceded by washing, and placing the gas under the pressure due to the head of water in the cup when up to the level mark.

Residual volume assumed to consist only of nitrogen.—As already stated, the residual volume after the removal of the carbonic acid, oxygen, and carbonic oxide is tabulated as the volume of nitrogen. Small proportions of hydrogen are sometimes included with the nitrogen,

but it is usually unnecessary to complicate the operation by including tests for hydrogen. Such a test, and also alternative tests for the leading components, are described in *Engineering*, vol. l. pp. 383, 413.

Statement and verification of results.—The results of analysis are most conveniently and generally useful when stated by volume, the chief reason for this being the facility for comparison as to the proportion of air supplied. The volume of carbonic acid produced is precisely equal to that of the oxygen which enters into its composition. If then a quantity of pure carbon were burnt in air to the production of carbonic acid, the mixed gas produced would contain precisely the original volume of nitrogen, but carbonic acid would replace an equal volume of oxygen. If air be supplied in excess the same conditions would apply, with the exception that the replacement of oxygen would be only partial, but the sum of the volumes of carbonic acid and oxygen would still be equal to the original volume of oxygen contained in the air. The percentage composition of the atmosphere is 20·81 of oxygen and 79·19 of nitrogen, with other substances in minute proportions. Consequently in all such simple cases as the one above given, the proportion of nitrogen in the mixed gases equals that of carbonic acid—or that of carbonic acid *plus* that of oxygen—$\times$ 3·805. But in many cases appreciable proportions of carbonic oxide are produced. This gas occupies double the volume of the oxygen which enters into its composition; consequently the amount of nitrogen corresponding to the oxygen consumed in its production equals the quantity of carbonic oxide $\times$ 1·902. Hydrogen also enters into the composition of all fuels, and in its combustion produces water, which is condensed from the furnace gases when their temperature is reduced before analysis. This

water, therefore, does not appear in the statement as to the volumes of the several gases obtained in analysis. But a certain amount of oxygen has disappeared in the production of the water, and the nitrogen, which together with this oxygen formed the air supplied, remains in the gases to increase the proportion of this gas. The proportion of oxygen which thus disappears under normal conditions amounts to

$$\frac{3\left(\%\text{ hydrogen in fuel} - \frac{\%\text{oxygen in fuel}}{8}\right) \times \left(\%\text{ carbonic acid in gases} + \frac{\%\text{ carbonic acid}}{2}\right)}{\text{percentage of carbon in fuel.}}$$

If this percentage, that of carbonic acid, that of free oxygen, and half that of carbonic oxide, are added together and multiplied by 3·805, the product should nearly agree with the percentage of nitrogen as obtained in analysis. In most cases the proportion of nitrogen should be about 80 per cent., but it is increased to about 85 when liquid fuel and others containing much hydrogen are used. Naturally, when air in large excess is admitted, the proportion of nitrogen in the waste gases tends to approach that in atmospheric air. If the average coal, of which the composition is given on a previous page, is burnt in 100 per cent. excess of air, and with a small production of carbonic oxide, the furnace gases should exhibit about the following composition—

Carbonic acid ...	8·96
Oxygen	10·53
Carbonic oxide ...	·37
Nitrogen ...	80·14
	100·00

If, on the completion of an analysis, the figures show more than 81 per cent. of nitrogen, or very much more than such as accords with the check just given, while

the fuel contains only a moderate amount of hydrogen, there is some probability that the gases have been irregularly collected, so that those from a freshly-stoked fire preponderate. Stale pyrogallic acid or potash will fail to remove all the carbonic acid, or oxygen, or both. Errors in either direction will arise if leakage into or out of the burette is allowed, even if apparently minute. The occurrence of carbonic oxide will reduce the proportion of nitrogen. Owing to differences in density between the several gases, the relative figures are very different when stated by weight from those of volume, therefore in either case the method should be positively stated.

Preliminary practice.—For practice, or for testing the chemicals used, an analysis of air may be made, as the composition of this is known. A sample may also be drawn from the flame of a Bunsen burner by means of an inverted tobacco pipe. By restricting the supply of air, as much as 5 per cent. of carbonic oxide may be produced, which gives a better sample for practice than ordinary flue gas. Owing to the large amount of hydrogen contained in coal gas, the quantity of nitrogen in a sample obtained from the combustion of coal gas is large.

NOTES.

Place all vessels, with the samples of gas, sufficient water, and the re-agents, in the room where the analysis will be made. This should be done in time for the whole to acquire the temperature of the room, and the whole should be placed beyond the reach of direct heat from the fire or the rays of the sun.

Never hold the burette in such a manner as to allow the transmission of heat from the hand or body to the gas in the instrument.

Close the cocks carefully but firmly, so that they will not drop out upon a slight touch. Hold the instrument in such a manner as to avoid moving the cocks.

Shut the cocks close at once upon the completion of the purpose for which they were opened.

Never remove or insert the stopper of the cup except when the cock is closed.

After due deliberation open cocks gently, and partially close in good time.

Cocks may be lubricated so that they will work steadily by means of a thin film of heavy mineral oil or vaseline. Organic oil or grease used for this purpose is partially removed by the potash solution, and sometimes causes slight leakage.

Before reading off a volume which affects the final analysis, the burette must be placed in communication with the cup, with the water in the latter accurately to its proper level.

Allow the liquid to drain from the sides of the burette always the same length of time before taking important readings.

Better commence with an odd volume than neglect accuracy.

Always use tube of small bore, and use glass where possible.

Keep aspirator bottle well supplied with water, and keep all tubes free from air when in use.

Keep all bottles containing solid or liquid re-agents well stoppered, and in a shady, cool place.

CHAPTER XI.

WATER FOR USE IN BOILERS.

Impurities in water.—The amount of solid matter dissolved or suspended in the water supplied to a boiler has a most important effect upon its efficient working. Pure water very slightly aerated is an ideal substance for the purpose, and is often provided by means of separate condensers. In ordinary practice, rain-water collected from ground free from limestone formations is the purest and best for use. Even this, however, is contaminated by acids and other impurities from various sources. But the bulk of water used in boilers contains varying quantities of mineral matters in solution or suspension. In a limestone district, practically all water collected from the ground contains much carbonate of lime, sulphate of lime, carbonate of magnesia, and smaller quantities of other substances, which may be neglected. In districts where limestone does not predominate, there is usually but little lime contained in the surface-water. But the water from deep wells often contains lime, in districts where the quantity of lime in the ground is too small to seriously affect the water in the streams.

Solid matter accumulates in boiler.—Practically nothing is evaporated in the boiler except water and

carbonic acid, all other impurities remaining in the boiler after the evaporation of the water which held them in solution. This solid matter is deposited on the plates of the boiler, interfering with the transfer of heat from the furnace gases to the water in the boiler, which causes waste of heat, and in advanced cases damage or ruin to the boiler by overheating. This deposit occurs most abundantly on the furnace tubes and water tubes where the heat is greatest, but some is found in other parts.

Hardness of water due to impurity.—Water containing lime and magnesia is said to be hard, the term probably arising from the sensation of harshness, and the difficulty in obtaining a soapy lather, experienced in the use of such water for washing. That which contains carbonates is said to possess temporary hardness; and that which contains sulphate of lime is said to possess permanent hardness. This distinction refers to the comparative difficulty attending the removal of the several substances. Carbonate of lime is practically insoluble in pure water; but if the water contains carbonic acid in solution it is enabled to dissolve about 85 grains per gallon. Though this appears to be a small quantity, it is sufficient to render the water quite unfit for use in boilers. Water containing 18 grains is, however, very largely used, and assuming that a boiler evaporates 8½ pounds of such water per pound of coal consumed, it follows that the water evaporated by one ton of coal will leave in the boiler or its connections 5 pounds of solid matter.

Precipitation of carbonates by suppression of free carbonic acid.—Since the solubility of carbonate of lime depends upon the presence of free carbonic acid in solution, it is obvious that the removal or neutralization of the latter will be followed by the precipitation of the former. The carbonic acid may be driven off by simple

boiling. Quicklime—in the form of lime-water, which is a clear solution—will combine with the carbonic acid to form carbonate of lime, which is precipitated along with the original carbonate, leaving the water pure, with the exception of 2 or 3 grains of carbonate of lime per gallon, which are soluble in pure water free from carbonic acid. The same object may also be secured by the addition of caustic soda, which combines with the carbonic acid to form carbonate of soda, which remains in solution, while the original carbonate of lime is precipitated. Carbonate of magnesia is held in solution in the same manner as carbonate of lime, and may be removed by the same means. The precipitate of one or both carbonates collects on the plates of a boiler as a fine powder, which is quite loose unless baked by the hot plates when the boiler is emptied before being allowed to cool down.

Removal of sulphate of lime.—Sulphate of lime is soluble in cold water, but is precipitated on heating to the temperature of the water in a high-pressure boiler, and forms a hard, troublesome scale, practically identical with gypsum or plaster of Paris. The addition of carbonate of soda causes precipitation of carbonate of lime, leaving sulphate of soda in solution. This precipitation arises from the transfer of carbonic acid from the carbonate of soda to the lime, forming insoluble carbonate of lime. In this reaction, caustic soda or quicklime, which do not contain carbonic acid, are of no direct use, and they accumulate in the boiler when thus misapplied. An accumulation of soda in the boiler causes priming and irregular working, while quicklime forms a hard scale upon the plates of the boiler. Either caustic soda or quicklime may, however, be beneficially employed in moderation to neutralize any free carbonic acid which may be present in the water, as, if this is allowed to

remain, it will (according to its amount) dissolve the whole or part of the carbonate of lime as quickly as it is formed, and thus frustrate the process of purification.

Restrictions upon use of soda.—In processes for the purification of water for use in boilers, it is desirable to avoid the use of soda compounds, except in cases and in amounts absolutely necessary, for the reason that the whole of the soda supplied in any form remains in solution as sulphate or carbonate of soda, or as caustic soda. When these substances accumulate in the boiler, they give rise to disturbance and pounding in ebullition. All soda compounds, but especially caustic soda, give rise to corrosive action on brass fittings, copper being dissolved. The dissolved copper promotes, if it does not actually cause corrosion and pitting of the iron in the boiler. The water in the boiler should therefore be changed freely in such cases. All saline matter in a boiler promotes galvanic action, which usually has a destructive effect. But when means of galvanic protection are adopted with a view to the prevention of corrosion, a slightly saline condition of the water is advantageous.

Use of lime.—In water purification, as much lime should be employed as will produce any effect, and soda to complete the operation up to the limits desired. The hardness of water is described by degrees which give the number of grains of carbonate of lime per gallon, other impurities being included in equivalent proportion. If the impurity consists entirely of carbonate of lime and magnesia, it may be removed by the use of two ounces of quicklime per 1000 gallons of water treated per degree of hardness removed from the water. Thus if 100,000 gallons of water are to be reduced from 21 degrees to 3 degrees of hardness, 225 pounds of lime will be required: 106 grains of pure

carbonate of soda will suffice to precipitate 136 grains of sulphate of lime. But carbonate of soda often contains considerable quantities of combined water, which adds to its weight, without adding to its utility. Bicarbonate of soda should not be used, as it yields an excess of carbonic acid, which may interfere with the free precipitation of the carbonate of lime. The materials are mixed with water in the required proportions by hand on a small scale, and on a large scale by means of mechanical agitators. After allowing to subside so as to give clear solution, or otherwise cream of the necessary consistency, it is mixed with the untreated water in the required proportions, by allowing each to pass through a nozzle of definite diameter into a mixing trough, in each case the head of water being maintained constant by a ball-cock or other means.

Double precipitation by use of soda in an exceptional case.—If the proportions of carbonates and sulphate of lime are nearly equal, the free carbonic acid, which causes the solution of the carbonate of lime, may be neutralized by caustic soda, forming carbonate of soda, which will react on the sulphate of lime, and so cause a double precipitation by the use of a moderate amount of one re-agent.

Applications of the Clark process of lime precipitation and filtration.—The original process of water purification by lime precipitation was suggested and practised by Dr. Clark, who mixed the water with the necessary proportion of lime, and allowed the whole to become clear by subsidence in large tanks. This is efficient on a comparatively small scale, but on a large scale is too costly. In the Porter-Clark system, filters are adopted for the more convenient and cheap separation of the precipitated matter. In these filters, the turbid water is passed through cloths stretched on frames, which are

bolted together in such a manner as to provide a large surface of filtering material in little space. The filtration is most perfectly effected when the cloths are covered by a thin coating of precipitate. When the precipitate accumulates, so as to interfere with freedom of filtration, the filter-cloths are removed and washed. Messrs. Atkins adopt filters in which the medium is also cloth, but applied to circular discs, the washing process being effected by revolving the series, and sprinkling water upon each cloth from a perforated pipe. Steam is occasionally blown through the cloths when it is found that the filtering power is not completely restored by water washing. In the Stanhope purifier, a series of plates are set after the manner of herring-bone walling, with very small spaces, through which the water is passed in an upward direction. On these plates the precipitate is deposited while the vessel is full of water. When this has proceeded a sufficient length of time, the whole of the water is drawn off, bringing the precipitate with it in a most convenient manner. This system is an excellent one for many purposes, but a very faint turbidity is apt to remain. This is, however, not accompanied by any trouble, beyond the occasional blocking of the cones of injectors, when these are used. The softening action takes place with greatest facility at a temperature of 150° F. It is also promoted by the addition of a little alum, which possesses some power of removing matter from suspension. Mechanically suspended matter may also be more or less completely removed by the several systems of deposition or filtration.

Effects of softened water supplied to a boiler.—It is found that when softened water is supplied to a boiler which has previously become coated with scale, the scale will drop off in pieces after the lapse of a few

weeks. This probably arises from the formation of fine cracks, from the difference in expansion between the iron and the scale. These are probably formed in a similar manner in ordinary work, but are as quickly filled up and sealed by fresh deposit, as may be seen on a broken surface of scale. Possibly the soft water may increase the shedding by exerting a solvent action upon the old scale. In any case, the plates, when once cleaned, will remain quite free from deposit so long as the supply of soft water is maintained and a moderate amount of blowing-off is practised.

Salt in water.—Common salt is occasionally present in appreciable quantities in stationary boilers, when the water is obtained from brackish creeks or from a salt district. Its presence is objectionable only in the same degree as that of other soda salts. Salt cannot be removed by chemical means, and blowing-off is resorted to. The water must never be allowed to exceed $\frac{3}{32}$ density as shown on the salinometer, equal to a sp. gr. of 1·081, or to a composition of 3 of salt to 32 parts of water. The density should, however, be usually kept much below the point named. When water is blown off to reduce the density, a large proportion of any carbonate which it may contain is simultaneously disposed of. But most of the sulphate of lime is retained as a very hard scale in the boiler. For this reason, blowing-off should be adopted very sparingly when sulphate of lime occurs largely in the water.

Acidity in water.—Acid matter is often found in water, derived from the sulphurous acid in the atmosphere arising from the combustion of coal; from decomposition of peaty and other organic matter; from chemical works; and from the water discharged from mines. This causes corrosion to proceed with rapidity

proportionate to the amount of acid present. It is practically impossible to remove such substances, but they may be neutralized by the application of caustic soda or carbonate of soda. An excellent and delicate guide as to the presence or absence of acidity in a boiler is furnished by the use of about two ounces of phenol-phthaleine in each boiler. This gives a faint pink colour to the water which is seen in the gauge glass. When the colour vanishes, or becomes weakened, the water has assumed an acid condition, or has lost much of the re-agent by blowing-off or by leakage at the cocks.

Mixture of waters.—Acidity—except such as arises from the dissolved carbonic acid—does not occur in water which contains carbonates of lime and magnesia, as these substances would neutralize other acids, and hence the comparative safety of such waters as to corrosion and pitting of the boiler. In some cases waters of two different natures are available for simultaneous use, and can be arranged so as to counteract the objectionable features of each other.

Decomposition of magnesia salts.—Chloride and nitrate of magnesia are decomposed at high temperatures, to the production of acids, which are most destructive if not promptly neutralized. If the carbonates are in excess the acidity is overpowered, but treatment by soda and frequent blowing-off are more effective.

Application of soda.—Soda for the neutralization of acidity is admitted directly to the boiler. This should be done in small doses, as any appreciable excess at one time leads to priming and violent ebullition.

Undisclosed compositions.—Compositions innumerable are made and injected into the boiler with a view to precipitate the impurities in the bottom of the boiler, in such a form as to be easily blown out. Some of

these are of organic nature, and appear to change the physical character of the matter precipitated. The best of such compositions are, however, those which in their action approach most nearly to the simple lime and soda treatment separately from the boiler, while treatment in the boiler is most clearly disadvantageous. For efficiency, even the best of such compositions must be regularly supplied in considerable quantities, and as regularly removed by blowing-out.

Testing of feed water.—Any person with very little chemical knowledge may test feed water, and direct the work of purification with complete success. An accurate analysis can, however, only be made by a chemist who possesses special knowledge and appliances for the purpose. A sample of water collected for analysis should be not less than two gallons in quantity, and should be carefully taken as an average sample. The jar and cork should be perfectly clean, and several times rinsed in the same water. The cork should be pressed in level with the neck, or a little below, and sealed. If carbonate of soda is used in the treatment of the water, it should also be tested, and arrangements made to secure a supply of this reasonably uniform in quality. The chemist's report will state the quantities of the several materials to be used in the treatment of the water, and will very much facilitate operations. But most waters vary somewhat at different seasons, and slight changes in treatment become necessary. The attention and trouble necessary in connection with water purification operations are not very great, and experience shows a most ample return.

Impurity due to the use of condensed water.—When water is used for boiler feeding which has passed through the engine—whether a jet condenser or a surface condenser is used—it is almost invariably pol-

luted by the material used to lubricate the piston, piston-rod, valve-spindle, &c., and with the slight amount of material abraded from their surfaces. The whole of the oil or grease used for this purpose must pass forward through the condenser and air-pump, with the slight exception that a small proportion is deposited in different parts, to be removed when the engine is opened. Lubricating materials of organic origin are very much changed in the passage, but nothing is destroyed. In such changes fatty acids are liberated, which have a corrosive action upon all metallic surfaces—especially those of iron—with which they come into contact. Pure, heavy mineral oils are free from liability to change, while they possess fair lubricating properties, and they should be used in all positions which may lead to their entry into the boiler. Moderate quantities of organic oils suffice to cause serious trouble by reason of accumulation of acidity in the boiler. This may be kept within bounds by careful attention to blowing-down, and it may be neutralized by the use of soda. But it is much better to avoid the cause of the trouble.

Dangers due to the presence of oleaginous matter in boiler.—The presence of pure mineral oils—whether changed or unchanged in their nature—in the boiler is not attended with the dangers due to acidity, and its consequent action upon the boiler. But the globules of mineral oil are found to adhere to any particles of limey matter which may be present, and especially in surface scum. The oil alone tends to float upon the surface of the water, by reason of its low specific gravity. The limey matter is of greater specific gravity, and would sink if in mass; but it is kept up by reason of its condition of minute sub-division. When, however, it becomes collected into particles of substantial size,

the effect of its superior specific gravity comes into action to overpower the buoyancy of the *oil*, and to cause the combined mass in the first instance to remain suspended in the water, indifferently in any position, and when further developed to sink to the bottom. The sinking power is, however, never very prominently developed, and the particles become attached to any part of the boiler beneath the water-line, every part being covered more or less thickly by a film which is of a nature very distinctly greasy to the touch. This may not appear to be so abundantly developed on the furnace tubes as elsewhere, chiefly for the reason that the oily parts are removed by the continued action of heat. A mixed film of oil and calcareous matter is exceedingly prone to cause dangerous overheating of the metal, by reason of its low conductivity for heat; and even a trace of grease on the surface too thin to measure, or to possess either positive or negative powers as to conduction, has a distinctly appreciable, prejudicial effect upon transmission of heat.

Prevention of grease in boilers.—When a boiler is found to be tainted in this way, no time should be lost in removing the greasiness and eradicating the cause. The first step is to ascertain whether any reduction can be made in the amount of oil consumed. The many forms of sight-feed lubricators now in use give great facilities for nicety of adjustment in this respect; in fact, they are indispensable for safe and efficient work. If the quantity of oil supplied cannot be materially reduced, it may be possible to partially change the water, so that one fractional part of the feed is fresh water, and the balance is taken from the condenser overflow. In some cases—perhaps the majority—in which this trouble is experienced, relief may be obtained by a judicious use of scumming apparatus, and

probably this may be the most important service which can be rendered by scumming apparatus. A large separator may be used, in which the water is slowly passed under and over a series of division plates. In such an apparatus the oil rises to the surface and the mud settles to the bottom, suitable means being provided for the removal of each as they accumulate. The most perfect appliances which are adopted for the removal of oil are filters. The most varied materials are used as filtering media—blankets, sponge, coke, and hay perhaps more than any others. The casing of the filter is almost invariably closed, so that the water may be carried through the material either by the suction or the pressure of the feed pump. The power of the suction is limited, and usually the filter is placed on the pressure side of the pump, though there need not be more than five pounds per square inch of difference in pressure between the two sides of the filter. Large masses of sponge may be used when the filter is required to work unopened for long periods. Hay may be used for shorter periods. Sponge may be washed and purified, or the hay removed and replaced by new. Woollen cloths or blankets are used in boxes or presses of different patterns. They are placed over gratings, or wire-netting, or gauze well supported and made water-tight all around, so that all water must pass through the filtering medium. The strength of the apparatus must be sufficient to withstand the working pressure. All covers and details which it may be necessary to deal with in opening out for cleaning should be freely accessible, and conveniently and rapidly taken apart and replaced. The provision of a spare set of filter-plates for cleaning while the alternative set is at work is a great convenience. Probably a number of perforated vertical cylinders, each with a turned joint at each

end, and covered with a cloth stocking well secured at each end, will prove most successful. Whatever description of filter is adopted, it should be remembered that frequent cleaning out of the refuse is necessary, or the object will quite fail of accomplishment. When a filter is kept in good working order the removal is effected of metallic particles and other bodies which cause damage to the boiler when present. The adoption of filters is of greatest importance in connection with surface condensation, but is useful in all cases where oil would otherwise enter the boiler, even though the quantity of this appears to be insufficient to cause trouble. In case of a jet condenser supplied from a cooling pond to which the water is returned, the adoption of a filter will keep the pond in sweeter condition.

References.—A paper by Mr. W. W. F. Pullen, published in the *Minutes of Proceedings of the Institution of Civil Engineers*, vol. xcvii. p. 354, gives an account of plant on the Porter-Clark principle for the purification of water for the use of locomotives at Penarth. A paper by Mr. W. Matthews, published in the *Proceedings of the Institution of Civil Engineers*, vol. cviii. p. 285, describes plant on the Atkins-Clark principle for the purification of the public water supply at Southampton. A paper contributed by Professor Lewes to the Institution of Naval Architects in 1891, and one contributed by Mr. Sylvester to the Birmingham Association of Mechanical Engineers, deal generally with water purification. A paper contributed by Mr. Edmiston to the North-East Coast Institution of Engineers and Shipbuilders in 1892 describes filters. A filter is also illustrated and described in *Engineering*, vol. lvi. pp. 47, 50.

CHAPTER XII.

BOILERS.

Preliminary considerations.—The first element to be regarded—after safety—in the consideration of an installation of boilers, is the amount of water to be evaporated. If the boilers are intended to replace existing ones, and especially if the engines are intended to remain unchanged, the quantity of water evaporated can be directly measured. In other cases, an estimate may be made as to the number of indicated horse-power, and multiplied by an assumed figure to give the total weight of steam required per hour. The consumption of steam per hour is found to vary from 13 pounds per indicated horse-power in well-designed triple expansion engines working at a pressure of 160 pounds above the atmosphere, to 20 or 24 pounds in compound engines of moderately good design, working at 60 pounds pressure. Any estimate based upon the indicated horse-power required must be subject to allowance on account of any steam required for purposes other than driving the engines.

Variation in heating surface.—Within the limits adopted in practice, an increase in the amount of water evaporated per pound of fuel is always secured by an increase in the heating surface of the boiler. Beyond

certain limits, however—which vary with the conditions —such increased evaporation is quite unremunerative, owing to the increase in the cost of the boiler, which becomes necessary, and the rapidly diminishing increments of heating effect, due to successive increases of surface. The limits of fluctuation in power to be observed, and the proportionate length of time during which low, mean, or high pressure is required, are important elements to be considered.

Variation in grate surface.—The grate surface must be sufficient to allow the consumption of fuel in sufficient quantity to generate the amount of heat required. The amount of coal which may be burnt per square foot of grate surface in ordinary practice depends upon the strength of chimney draft, as explained in the chapter on chimneys. The amount of heat to be obtained by the combustion of one pound of coal is dealt with in the chapter on fuel.

Proportions of heating surface.—The heating surface should just suffice for the attainment of the desired degree of economy in the utilization of the heat which is generated in the furnaces. In Lancashire boilers of ordinary proportions the area of heating surface is from 25 to 30 times greater than the area of grate surface. Good results are obtained when the amount of heat absorbed by the water in the boiler amounts to 6000 to 7000 units per square foot. Such boilers are usually provided with economizers to perform a preliminary heating of the water, but which must not be arranged with a view to the actual evaporation of water into steam. As explained in another chapter, the heating surface in an economizer is less powerful than that in the boiler, whereby the average is reduced.

Number of boilers adopted.—The number of boilers over which a given amount of heating surface should be

distributed depends largely upon the character of the fluctuations which may be met with in the amount of power required. Also upon whether the provision—always desirable—of a spare boiler may be possible.

Lancashire and Cornish boilers.—In the majority of cases, plain cylindrical boilers with internal furnaces have proved themselves to be distinctly superior to all others for stationary work. In large sizes two furnaces are used, and the type is known as the "Lancashire" boiler. The "Cornish" type contains only one internal furnace, and is generally adopted for small powers, or to provide close gradations in power required at different times. This type was largely used in connection with the Cornish pumping engines, in the early days of high-pressure steam.

Dimensions and pressures in boilers.—The majority of stationary boilers in current practice are of the Lancashire type, about 30 feet long, and from 7 ft. 6 in. to 8 feet 6 in. in diameter. To suit compound engines, the steam is supplied at a pressure of about 100 to 120 pounds above the atmosphere, for triple expansion engines 150 to 160, and for quadruple expansion engines at 200 pounds pressure.

Material.—Steel made by an acid process is employed in boiler construction almost to the exclusion of iron. The material used for the shell and ends gives a breaking test of not less than 26 nor more than 30 tons per square inch of the original area, and an elongation of at least 20 per cent. over a length of 8 inches. The material for the furnaces, and for such other parts as require to be welded, gives a breaking test of not less than 24 nor more than 28 tons, with 20 per cent. elongation, which usually, however, reaches 25 to 30 per cent.

Strength and arrangement of riveting.—The boiler shell is made in 8 to 10 rings, each ring being in one

plate of uniform thickness throughout. For *high* pressures, the longitudinal joints are butted with cover-plates inside and outside, with three rows of rivets on each side of the joint. The proportions adopted are such as to secure the closest possible approach to the full strength of the plate, and thereby avoid the necessity for excessive thickness throughout the shell. The strength of the joint depends chiefly upon the strength of the rivets as opposed to shearing, and upon the amount of plate remaining after the insertion of the rivet-holes. In a joint held by a single line of rivets, as in Fig. 5, each of these elements of strength may be increased at the cost of the other, and the whole so

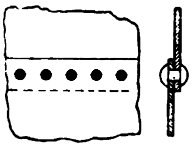

Fig. 5.—Single-riveted overlap joint.

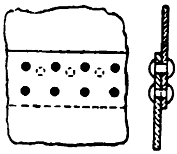

Fig. 6.—Double-riveted overlap joint.

adjusted that the two are practically equal. In such a joint the rivets may, however, be reduced to one-half the strength of the plate, by which means the strength of the plate will be increased to an important extent, though far from becoming twice as great as before. A second row of rivets may then be provided, as in Fig. 6, to restore the total strength of rivets, and the whole again adjusted to equality, when it will be found that an important accession of strength has been secured above that due to a single line of riveting. The second row of rivets may be arranged as shown in full in the figure, when the joint is said to be "chain riveted," or as shown dotted, when it is said to be "zigzag riveted." If instead of arranging overlap joints, the two ends which

are joined are brought into contact, end to end, or butted, they may be secured together by means of cover-plates or straps on one or both sides, held by rivets passed through the two or three thicknesses. In the latter case, as in Fig. 7, each rivet is placed in "double shear," so that it opposes rupture upon double area, and contributes strength accordingly; its strength is also applied in a more direct and efficient manner than when in single shear. The diameter of rivet necessary is therefore reduced, the strength of plate remaining after perforation for rivets is further increased, and

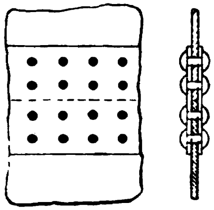

Fig. 7.—Double-riveted butt-joint, with double cover-plates.

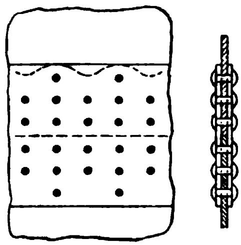

Fig. 8.—Treble-riveted butt-joint, with double cover-plates.

the rivets are more easily and efficiently closed up than they would be if of larger diameter. This advantage, however, assumes greatest importance in connection with boilers of the large diameters necessary in marine practice, and in which thick plates must be used. In boilers intended to work at very high pressures, a third row is added, as in Fig. 8, and sometimes a fourth row of rivets, so that the strength of the joint shall approach still more closely to the full strength of the plate. These rows are pitched at greater distances apart, with a view to leaving a larger proportion of the plate available for strength. The strength of the butt-plates must be at least equal to that of the shell, and in some

cases the thickness of the outer cover-plate must be increased still further, or other means adopted to obviate springing the same by caulking, which difficulty increases with any increase in the pitch of rivets in the outer row. For this reason cover-plates with indented edges, as shown dotted in Fig. 8, have been adopted to a limited extent in marine practice. A form of joint known as Rowe's patent has lately met with some favour. In this the cover-plates on the caulking side are made to take one row of riveting less at each edge than those on the opposite side, whereby the springing of the edge is reduced. As a rule, double-riveted joints with inside and outside cover-plates are made of strength equal to not less than 70 per cent. of that of the unreduced plate, and treble-riveted ones of 80 per cent. The longitudinal joints of all high-pressure boilers are made in accordance with one of these, the cover-plates being cut with the grain in the same direction as the plates—*i. e.* circumferentially disposed—and well fitted all over. The several rings of the shell are parallel, and are arranged alternately larger and smaller, so as to fit inside and outside of each other, the circumferential seams being overlap joints double riveted for high pressures. The longitudinal joints are placed alternately on the right and left sides of the boiler, but in all cases above the side-flues, and clear of the gusset-stay angle-bars.

Use of small plates.—In older practice, each ring of the shell was composed of two, three, or four plates. A boiler thus made possessed many weak points as to direct strength and as to variable elasticity, each ready to yield by reason of stress due to heat expansion, or changes in weight due to filling or emptying the boiler, and other causes. Such tendency is magnified by irregularity in the seating and by vibration set up by

machinery in close proximity. A boiler made in this way is certainly much inferior to one made with each ring in one plate. If constructed in a well-equipped boiler-yard, it will cost more than one in which each ring consists of a single plate, though the small plates are more easily manipulated.

End-plates.—The front end-plate is secured to the shell by an outside angle-ring, prepared for double riveting. The back end-plate is flanged to fit inside the shell, and secured by double riveting. Each of these plates is in one piece, and must be secured in such a manner as to receive support upon the flat surfaces, while a sufficient amount of elasticity is allowed for the expansion and contraction of the furnace-tubes. A large number of Lancashire boilers have been provided with two longitudinal stay-bolts, reaching from end to end of the boiler, and provided with nuts and washers, fitted about in the centre of the flat surfaces which exist above the furnace-tubes at each end of the boiler. Probably these bolts are efficient, provided that they are originally well fitted and not afterwards disturbed. These conditions are, however, not always regarded, and on this account some leading authorities prefer to rely on well-proportioned gusset-staying only, as shown in Fig. 9, secured to the shell and to the end-plates by double angle-bars. In the absence of bolts, five gusset-stays are usually provided at each end, above the furnaces, and three at back and two at front end below the furnaces, one being in the latter case displaced by the mud-hole. The lower gussets are all attached to the end-rings of the shell; but the upper gussets should be as far as possible alternately attached to the end-ring and the second ring at each end. These gussets being securely attached to both shell and end-plates by strong angle-bars on

each side, it follows that the end-plates are practically immovable, relatively to the shell, so far as the gussets extend. Freedom for expansion over the furnaces is

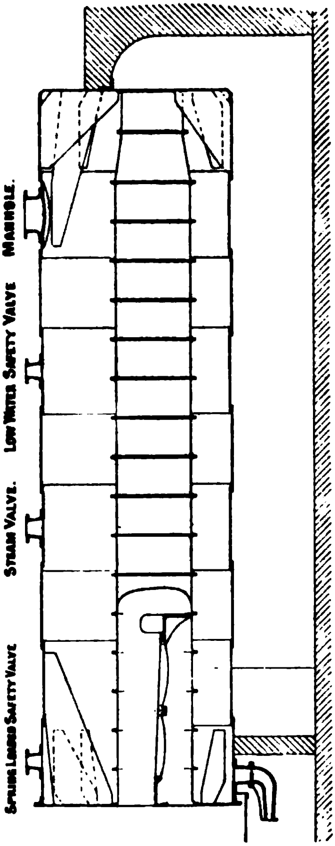

Fig. 9.—Lancashire boiler, longitudinal section.

secured by stopping the ends of the gusset-stay angles so that the lowest rivets are at a distance of 10 inches —measured centre to centre in the shortest direction—

above those which secure the flues to the end-plates. About 5 to 6 inches should be similarly allowed to the rivets of the lower gusset-stays. The elastic yielding

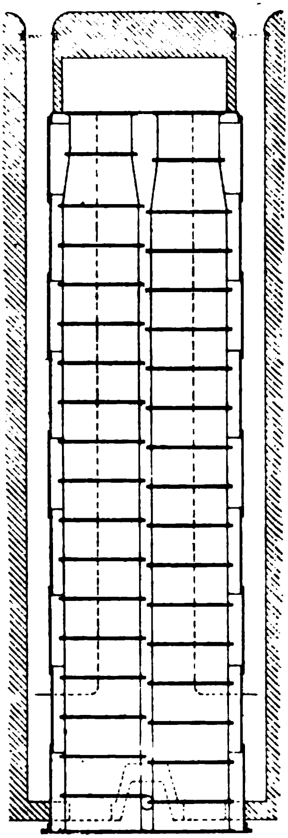

Fig. 10.—Lancashire boiler, sectional plan.

or work, as it may be termed, is distributed over the two end-plates and the intermediate joints in each internal flue, so that it has comparatively little effect

at each place. The riveting of the gusset angle-bars must be sufficiently openly pitched to avoid reduction

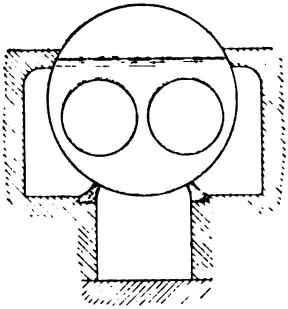

Fig. 11.—Lancashire boiler, cross-section.

of strength of shell, *i. e.* not more closely than the most open row in the longitudinal seams of the shell.

Internal flues.—For high pressures the internal flues are made in about 18 rings each. For a boiler of 8 feet diameter, the 16 rings next to the front are usually made of 3′ 2″ diameter, the 17th is tapered from 3′ 2″ to 2′ 7″, and the last one is 2′ 7″ throughout, for

Fig. 12.—Adamson's flanged joint for furnace-tubes.

convenience of access to the bottom inside. Each ring is welded longitudinally and flanged outwardly. The several rings are successively brought together, a thin annular plate or caulking-ring being inserted in each joint and riveted, as shown in Fig. 12. The caulking-ring serves to make the joint tight by fullering, without indenting the furnace-plates, and also imparts some strength against collapse. This is known as "Adamson's flanged joint," and avoids the exposure of

any rivet or more than a single thickness of plate to the action of the fire. The two furnace-tubes for one boiler are precisely alike, with the exception that the joints in one are kept about six inches further forward than in the other one to give clearance, as shown in Fig. 10. For the same reason, these joints should not be allowed to fall opposite the shell seams. They should also be so disposed that the strength of the flame in curving to pass over the bridge does not act upon a joint. The tubes are usually flanged at each end for attachment to the shell, as in the Adamson joint, but omitting the caulking-ring, the end-rings of the furnace-tubes being made of a greater thickness than the rest; or a strengthening ring is shrunk on the end and flanged together with it, giving virtually a thick edge to the plate; or the plate is originally rolled with a thick edge. Sometimes, however, the ends of the tubes are left plain or unflanged and secured by angle-rings, but it is doubtful whether this is as satisfactory as a flanged connection, giving less strength and elasticity, and exposing a riveted seam to the heat, which is objectionable at all times. Various other forms of joint for furnace-tubes have been introduced at different times, but are not extensively adopted. With a view to obtain great strength against collapse, without the use of excessively thick plates, the number of rings now adopted is greater than in practice a few years ago. Some additional amount of longitudinal elasticity is also secured by reason of the more numerous joints and thinner plates, as compared with those which would otherwise become necessary. The general arrangement of a Lancashire boiler with flues and seating is shown in Figs. 9, 10, and 11.

Galloway tubes.—About five taper, water-circulating tubes ("Galloway tubes") are usually inserted in each

flue, with a view to improve the water circulation, increase the amount of heating surface, and effect an improved distribution of the hot gases.

Galloway boilers.—The Galloway boiler is essentially a Lancashire boiler modified to give increased heating surface, or a given amount of heating surface in a shorter length. The modification in form consists in uniting the two furnace-tubes into one combustion-chamber of kidney-shaped section, Galloway tubes being provided to give the required heating surface. The boiler is set and used in the same manner as an ordinary Lancashire boiler, and is the only modified Lancashire boiler which has proved successful.

Treatment of material.—All plates should be bent cold, by means of rolls, and openings cut out by machine, using a cutting tool and not a punch for the purpose. All holes throughout are drilled. Throughout the boiler, all material which, either at the edges of plates or around openings large or small, has been in contact with a punch or with shearing knives is removed by planing machine, lathe, drill, or other cutting tool.

Drilling, riveting, and caulking.—In every important boiler-works at the present time all rivet-holes are pierced by drilling machinery, which has been brought to great perfection. By this means all possibility of molecular damage to the plates is avoided. The holes through the whole of the thickness—whether two, three, or four—which will be filled by any particular rivet, are drilled at one pass, the whole being tacked together in position for the purpose. The practical result of this is, that the holes fit each other with absolute truth before and after they are filled with the rivets. But before the rivets are inserted, the plates are taken apart, the slight burr raised by the drill is removed, and also the sharp edge, so that each side of each plate

is to a slight extent conically recessed, this being greater beneath each rivet-head than at the intermediate surfaces. The surfaces of the plates where they come into contact should be treated with a solution of sal-ammoniac, to remove all traces of mill-scale, and leave perfectly clean metal. Hydraulic or pneumatic machinery is used for closing the rivets, and the riveting machinery is so designed that exceedingly few rivets remain to be closed by hand. Rivets of best Yorkshire and other similar qualities of iron have been largely used, but rivets of soft tough steel are now used almost to the exclusion of all others. In ordinary practice of twenty or thirty years ago, the plates were not necessarily brought and held in close contact by the rivets, so that severe local caulking of seams was necessary for obtaining a water-tight condition. Rivet-holes were punched with a very rough degree of accuracy, and were faired by the violent use of drifts, often causing cracks before the boiler was put to work. In all works of repute, the plates are now well bedded together, so that the joint is almost tight by riveting, but is completed by fullering with a wide tool, which leaves very slight marks, and quite avoids the indentations to which plates were formerly subjected. Safety under the high pressures of the present day would have been practically impossible if the rough practice of former times had not been abandoned.

Flanging and annealing.—Flanging is now largely adopted in the attachment of plates, the surfaces of which intersect. The flanges for this purpose are turned at one heat by means of special machinery. In best work the back end-plates are also flanged at one heat around the circumference to fit the shell, small plates being flanged by dies at one operation, and larger ones in short lengths by means of anvil-

presses. The end-plates of marine boilers are also flanged to receive the furnaces. All plates which have been flanged, or which have in any way been worked under heat, should be subsequently carefully annealed. The material should also be exposed to suitable temperatures while under work. If the temperature should be allowed to rise too high at any time, the nature of the material suffers and cannot be restored. But if any work is done upon the material at a temperature below the suitable one, a local straining appears to be caused, which may be removed by annealing, unless the damage is extreme. If this process is omitted, there is some probability that the piece will crack across, sooner or later. Annealing, by which such strained condition can be entirely removed, consists in raising the whole of the piece to a uniform red heat, whereby the particles are allowed to adjust themselves to a condition of equilibrium, after which the plate is allowed to cool down uniformly and very slowly. The uniformly good quality of material now used, and the efficiency of the annealing process are such that defective plates occur only with extreme rarity.

Mounting-blocks or stand-pipes and covers.—Openings are required to be made in the shell of a boiler at various places to suit the several fittings. These openings cause a loss of strength in proportion to the amount of material removed. The requisite fittings can be applied much more conveniently, securely, and accurately to a rigid plane surface than to the curved surface of the boiler-shell. Mounting-blocks or stand-pipes of wrought-iron, or occasionally of cast-steel, are therefore attached to the shell at each opening of important size. These are attached by riveting, and therefore give a much more steam-tight joint than could be secured by the use of bolts or studs. Bolts

and studs which are exposed to steam-pressure at one end and to the atmosphere at the other end, are especially liable to leakage, which is objectionable in itself and which leads to rapid corrosion, so that each bolt becomes weakened and the surfaces pitted, great trouble being caused in both ways. The mounting-blocks more or less completely restore the lost strength of the boiler-shell, and furnish an excellent base for the attachment of each mounting by bolting, in such a manner as to allow of its convenient removal and replacement at any time. The opening for the man-hole, being the largest in the boiler, requires a further strengthening in addition to the one described, and this is furnished by a doubling-plate which is made larger than the flange of the block, and which is secured by the double riveting of the block, and also by an additional row of rivets beyond, which are pitched somewhat further apart. This doubling-plate may be either circular or rectangular in form. The man-hole is provided at its upper part with a flange for the reception of the cover, which is a plain plate secured by ordinary turned joint-bolts. The mud-hole at the lower part of the boiler-front is provided with a mounting-block standing inward; the cover is secured by two cross-bars, with a strong bolt in each. An outer cover is also provided for the sake of neatness. The man-hole usually gives a clear circular opening of 16 inches diameter; that of the mud-hole is oval, $16'' \times 12''$, or in exceptional cases $15'' \times 11''$. A mounting-block is provided to receive the main steam stop-valve. A second block of smaller size with corresponding stop-valve should be provided for subsidiary work, such as driving pumps. By this means the necessity for filling the main steam-pipe with steam, on many occasions when only a small quantity is required, is avoided.

Mounting-blocks are also required for each of the two safety-valves.

Steam stop-valve.—The main stop-valve is a screw-down valve, with the casing made of cast-iron for pressures up to 90 pounds per square inch, and of cast-steel for higher pressures. The valve-disc and seating are usually of gun-metal or hard bronze, but cast-iron may be adopted where not exposed to powerful corrosive action. This expands equally with the casing, which is a great advantage. The spindle should be of gun-metal or of wrought-iron covered with brass over the portion within the stuffing-box. The screw-thread should be outside the casing, for facility of inspection and lubrication. This is always heavily loaded, so that the area of screw-thread should be made as great as is practicable. The nut should be fitted in a wrought-iron cross-head, carried by two turned studs. In most cases the pressure is below the valve, so that the stuffing-box is free from pressure when the steam is shut off by the valve. A narrow width of seating—say one-sixteenth of an inch—is more easily kept tight than a greater one, where it can be practically adopted. But when the pressure is above the valve, the load upon the seating becomes so great that a narrow seating is soon destroyed. In all cases, a small auxiliary or bye pass-valve is of great convenience in carefully warming the pipes, and is also useful in relieving the load upon the main-valve before opening, by which means the operation is greatly facilitated. In some cases a supplementary valve is applied over the main-valve so that it is opened by the first turn of the screw, before the main-valve is opened. This is however only applicable when the steam is applied above the main-valve. The strength of the valve should be such that no appreciable deflection

shall occur in use. The proportions should be such that in no case will the valve lift more than one-fourth of its diameter. When open, in ordinary work, it should be screwed close up to the stop with a view to reduce vibration and wear. The thoroughfare area should be such as to allow the passage of the required amount of steam without its velocity at any point exceeding the corresponding velocity allowed through the steam and exhaust-ports of the engine, which is elsewhere treated. It is practically impossible to design the casing of a stop-valve without some irregularity of form, but this should be minimized. The dimensions of the whole should also be kept within moderate limits. In many cases stop-valves are situated at the termination of a straight run of pipe, and exposed to especial risk of rupture from shock caused by water in the pipes. Boiler stop-valves are comparatively little exposed to this risk, but all others are, and the same patterns are used throughout. The flanges of the stop-valves which fit against the seating-blocks or pipes should be of suitable width to receive bolts through both flanges to secure the joint. Brackets should as a rule be provided between each pair of bolts—or otherwise in alternate spaces—to give strength to the whole, and stiffness of flange to secure a tight joint.

Collection of steam.—A horizontal pipe, called an "anti-priming pipe," is usually provided inside the boiler for the purpose of collecting dry steam, or rather collecting the steam in such a manner as to avoid the passage of a rapid current over the water while disturbed by ebullition, by which unevaporated water is carried forward by the steam. This is best placed elsewhere than over the disturbed water above the furnaces, closed at each end, and perforated with slits. The anti-priming or collecting-pipe is always connected to the

main steam-valve. In some cases the same pipe may with advantage be connected to the auxiliary steam-valve.

Steam-receivers.—Steam-drums, receivers, and domes have in the past been largely adopted for the purpose of ensuring a supply of dry steam. In very few cases did these conform to the conditions set forth in the chapter on evaporation. They therefore usually failed to secure any advantage, while they interfered to a serious extent with the strength of the boiler. Even when correctly designed in other respects, they are of comparatively little value unless connected with an anti-priming pipe, or otherwise connected with the boiler at two or more points. There is an impression that by some means steam becomes drier in a dome. But everything which enters the dome leaves it by way of the steam-pipe, so that the steam cannot lose any of its contained moisture, while the exposed surface of the dome, whether clothed or unclothed, must cause some amount of condensation.

Cutting action of steam-currents.—In connection with mounting-blocks, anti-priming pipes, and indeed in all fittings exposed in any way to the current of steam, the cutting action of a current of live steam should always be kept in view. It is not necessary to discuss how far this may depend upon the presence of minute particles of liquid water, or of matter other than pure vapour, or whether promoted by corrosive action. Experience shows it to exist, and the arrangement of perforations in the anti-priming pipe, and of its attachment to the mounting-block or the shell of the boiler, should be so disposed as to avoid causing a high velocity of current in close proximity to the shell. It is also equally important that the currents of steam shall be so directed as to avoid direct impingement upon the surfaces.

Safety-valves in duplicate.—Two safety-valves should be provided, each of such capacity as to ensure complete relief of the boiler from over pressure when steaming at its normal rate, so that in case anything should arise, no danger tending to cause increase of pressure will be incurred.

Dead-weight safety valves.—For moderate pressures the dead-weight type of safety-valve has been largely and successfully used. With the adoption of high pressures, the necessary weight becomes objectionable, unless the size of the valve is reduced, which cannot be recommended.

Spring-loaded safety-valves.—During recent years great improvements have been made in the manufacture of spring-loaded safety-valves, and one such valve is very generally applied to each high-pressure stationary boiler. They are compact and reliable, do not subject the boiler to any shock, can be eased off conveniently to test their freedom, and quite prevent any serious rise of pressure. Other valves compare favourably in commencing to blow off with accuracy at loaded pressure, but few—if any—are superior in their ability to discharge large quantities of steam upon a moderate rise of pressure. A compound valve with dead load is said to answer well in this respect for moderate pressures, but for high pressures any system of dead loading is cumbersome.

Low-water safety-valves.—The second safety-valve is generally loaded by weight and lever, and is also arranged to discharge when the water falls below a level, determined with a margin above the lowest safe level. A float is submerged in the water at such a level that the floating power of the water is withdrawn when the water falls the permitted distance. The weight of the float acting downwards then operates upon

a vertical rod which opens a secondary safety-valve which is seated upon the main-valve, which is directly weighted by lever.

The secondary valve being relieved, the main-valve is more free to lift. But if it does not lift at once, the pressure in the boiler is relieved by the secondary valve and an alarm is raised, so that help can be given to restore the water in the boiler, for which there is ample time, if such an emergency arises during working hours. If it should arise from a leakage from the blow-off cock, or from any other similar reason at a time when work is suspended, there is likely to be still more time available, and the steaming power of the boiler will almost certainly be below the discharging power of the valve, so that there is even less possibility of serious results following. These valves are also arranged to act in the same way if the water is allowed to rise to a dangerously high level, which is of great value in some exceptional cases, especially when the boiler is fed by water of a pressure higher than that of the steam in the boiler.

Precautions against irregular loading of safety-valves.—It is of vital necessity that no possibility shall exist whereby any change can be made in the loading of a safety-valve, except when the boiler is out of steam. As far as possible, each valve should be locked, so that no person without complete authority can at any time make a change. Spring-loaded lever-valves must be ferruled, so that the balance cannot be screwed down too far, and weight-loaded lever-valves should have the weight locked in position, so that it cannot be moved along the lever, or removed and replaced by another one. Second weights are objectionable, owing to the opportunity for irregular increase of weight which they present.

Fusible plugs.—Fusible plugs which are placed in the top of each furnace-tube, and are intended to melt out when the water falls below its proper level, and admit steam into the furnace to extinguish the fire, are not found to be reliable for very high pressures. They sometimes melt out when they ought not to do, and fail to melt when the occasion arises. When fusible plugs are adopted, they ought to be carefully examined on every possible occasion, and the caps containing the fusible metal should be frequently exchanged for new ones. Preference should be given to those plugs in which the metal is disposed so as to leave openings of considerable size upon melting, whereby the liability to stoppage by incrustation is reduced.

Feed-pipes and valves.—The feed water is introduced into the boiler at the front end. A perforated pipe is provided, about half the length of the boiler, so as to distribute the water. The bottom of this pipe is set about three inches above the level of the furnace tops, so that if the feed-valve should become leaky, the furnaces cannot be laid dry in consequence. The feed adjustment-valve is very similar in arrangement to the main steam-valve, but is of smaller size; and the valve disc is arranged so that the raising of the spindle does not raise the valve, but only allows it to rise in case there is a greater pressure beneath than above the valve. This is with a view to its acting as a check or non-return valve, in case the necessity for such should arise. In some cases a separate check-valve is provided in addition to the one which can be closed as a stop-valve. This is however quite unnecessary, except where the check-valve is particularly subject to choking, in which case the upper stop-valve provides the means for examination of the check-valve at any time while the boiler remains under steam.

Where it can be conveniently arranged, the feed-pipes should be carried overhead rather than beneath the stoke-hole floor, whereby the pipes will be less subject to wasting and damage. The supply branch to each valve is then brought by a half-bend to the underside of the feed-valve, the valve itself being precisely the same as though the branch-pipe came up from the stoke-hole floor.

Scumming apparatus.—A scumming arrangement is often fitted, consisting of a line of receivers set inside the boiler accurately to the water level, and communicating with a plain cock outside, and thence by a pipe down through the stoke-hole floor, to the same waste-pipe as receives the water from the main blow-off cock. This arrangement is not universally required, and whether or not scum is removed by its use, the freshest and purest water is so removed. In cases where there is good reason for its use it often fails to receive proper attention, as to the level of water at the time of operation, or the time during which the operation is continued. The indiscriminate adoption of scumming apparatus therefore cannot be recommended. The question is treated generally in connection with feed water.

Blow-off cocks.—A blow-off cock is placed beneath the boiler and near to the front end. This is for use in emptying the boiler, and for the removal of sediment, or for blowing-off in connection with the admission of fresh water to the boiler. The usual type of cock adopted is a plug cock, asbestos packed, with a passage at the side to receive the water from the boiler, and one beneath to pass it off by the waste-pipe. The plug of the cock is arranged with its axis vertical, and with a square—cut at the top end to receive the key or spanner for turning the plug in opening

and closing the cock. This is arranged in connection with a guard, so that it shall be impossible to remove the key, except when the cock is accurately closed. In another pattern of cock made by Messrs. Hopkinson, a parallel double slide-valve is used, with the two faces of the movable valve held apart by a special spring, and the opening effected by means of a rack stem attached to the valve, actuated by a pinion, the spindle of which works through a stuffing-box. This is a well-made and successful appliance. All descriptions of blow-off valves should be made throughout in movable parts and casings of gun-metal or bronze of high quality, so as to minimize wear, and to maintain the rate of expansion approximately uniform throughout. In the past serious accidents have been caused by the adoption of iron casings and gun-metal inner parts. In blowing off, the entire valve becomes heated and the inner parts expand more than the outer ones, so that jamming is caused, and it has been impossible to close the valve or cock before the furnace crowns were laid bare. Jamming is caused in other cases by the adoption of a fine taper to the plug of the cock. This should never be less than one diameter in four lengths. In all cases, blow-off valves should be opened and closed several times when operated, being opened a little further each time. They should be alternately opened towards the right and the left hand, by which circularity is better preserved. The blow-off cock is always placed below the stoke-hole plates and a little forward of the boiler front, an elbow-pipe being interposed between it and the mounting-block, which is attached to the underside of the boiler, in the same way as those of the stop-valves and safety-valves are attached to the upper side. The elbow-pipe is arranged as a steel casting, tapering from a

large diameter at the upper end to the size of the blow-off valve—usually 2½ inches bore—at the lower end. The form is arranged with a view to strength, and to the provision of considerable capacity to receive the sediment which is formed in the boiler and which finally subsides here, and remains to be first blown out when the cock is opened. The elbow-pipe is strengthened by a deep rib along its upper side, so as to reduce the risk of breakage. With the same object, it is desirable that the waste-pipe leading from the blow-off cock should not be tightly bound in by brickwork.

Water-gauges.—The single cock at the top of the water-gauge and the double cock at the bottom are asbestos packed. The two branches are connected by a glass tube with soft packing. When the cock of the top branch and the upper cock of the lower branch are open, the water rises to the same level in the tube as in the boiler, and the position of the surface can be seen in the glass. Some boilers cause the water to foam and prime, and to show false indications in the gauge-glass, but this is seldom the case with the class of boiler under present consideration. In some exceptional cases the water-gauges are connected to the water-spaces by lengths of piping. In a boiler subject to priming, a short pipe used in this way is advantageous in steadying the water in the glass. Long pipes radiate heat, so that the water which they contain becomes cooled, and consequently more dense than the water in the boiler. In this way the water in the gauge may stand two or three inches below that in the boiler, which is an error on the side of safety, so far as the furnaces are concerned, but which leads to a reduction of steam space, and may cause extensive priming. In all best practice the cocks are packed

with asbestos, by which means they are kept tight a much longer time, and work more easily than with two gun-metal surfaces in contact. In best modern fittings, the mere rigid mass of metal also conduces to durability and success in working. The glass gauge-tubes often break, causing trouble and injury to persons near, and occasionally loss of life. Different measures have been adopted for automatic closing when a tube breaks, of which the best is probably the adoption of some form of valve, generally a spherical ball, placed in the stem of each branch in such a position that upon the breakage of the glass the outward rush of the steam and water carries each valve against the opening and stops the current. The successful application of this principle requires that there shall be no tendency for the valves to become fixed in their places by deposit, that they shall not become covered by a film of deposit of variable thickness, that they shall not interfere with the usual and constant clearing of the gauge by opening the cocks, and that they shall promptly return to their normal positions after the fitting of a new glass tube and the opening of the cocks into communication with the boiler, so as to again show the level of the water. Pistons have been adopted, which are in equilibrium so long as the gauge-glass is unbroken, and which allow the ordinary movement of the cocks, but from one side of which the pressure is withdrawn on the breaking of the gauge-glass, when the cocks are closed automatically.

Breakage of water-gauge tubes.—Gauge-glasses break from two causes, apart from damage by violence. The material is dissolved by the water, usually at a very slow rate, but more rapidly at high than at low pressures. The second cause is from strain caused by differences of temperature, more especially if suddenly

arising. Cracks are thus caused, which are immediately followed by the destruction of the glass. Composite tubes are now made at Jena, in which two thicknesses of glass with different co-efficients of expansion by heat are adopted. These are found very successful in withstanding changes of temperature. Probably other makers will also make improvements, but an automatic cut-off should never be omitted when high pressure is adopted. In all cases, the gauge-cocks should be closed when the boiler is left for any length of time. The water-gauges are provided in duplicate, so that the breakage of one tube does not interfere with the safe working of the boiler.

Pressure-gauges.—The pressure of steam in a boiler is shown by a dial gauge, with a pointer revolving in the same manner as one hand of a clock. In all the best gauges a curved metal tube of flat elliptical section is adopted. On the admission of pressure to such a tube, the sectional form tends to become more nearly circular, while the relative lengths along the inside and the outside of the curve are but little affected. The radius of curvature is increased and the tube straightens in proportion to the pressure. This movement is imparted to the hand by means of a toothed segment and pinion. The chief difference in the gauges of different makers is that some provide stronger and heavier movements than others whose gauges are more accurate and free from "back lash," and are reasonably durable under good treatment, but rapidly fail under such treatment as no gauge should be expected to stand. All steam pressure-gauges are provided with an inverted siphon, which contains condensed water, and which prevents the access of steam temperature, which would affect the accuracy of the reading, and gradually destroy the gauge. The original form of this was a plain bent

tube, but a pendant brass tube with a smaller one inside it is now usually adopted, which is neater and equally effective. In connection with each pressure-gauge, a nozzle should be provided to receive a steam-engine indicator for comparison of readings.

Furnace fittings.—Each furnace is provided with a dead-plate in front, which consists of a cast-iron plate, arranged to suit the furnace-door, and to carry the ends of the first range of fire-bars. In ordinary practice the fire-bars are made in two or three lengths. When made in two lengths, the cross-bars present less interference with the admission of air beneath the fire, but there is greater tendency for the bars to become distorted by reason of excessive heat of fire. The cross-bar or bearer which supports the ends of the fire-bars at the centre of the length of the grate is made of channel form, and the fire-bars are sometimes made with nibs or hooks to suit; so that while free to expand and contract, or to move slightly in a longitudinal direction, they are quite safe against dislodgment. The extreme ends of each range of fire-bars are usually inclined, so as to give the utmost possible freedom for expansion, and thus obviate one great cause for distortion when exposed to very high temperatures. The fire-bars are made of a tapering cross-section, deeper in the middle of the length, to impart strength, and with the ends increased in width, so that when in position they must give a definite width of opening—usually about half-an-inch—over the length of the bar, for the admission of air to the fire. This opening is carried parallel downwards about three-quarters of an inch from the top, so that the width may remain constant, though the upper surface of the bars is wasted by the heat. At this part the bars are $\frac{3}{4}''$ or $\frac{7}{8}''$ thick, from which they taper so that at the lower edge they are not more than half the thickness. The

after-end of the last range of fire-bars is carried by the bridge casting, which is also arranged to carry the bridge blocking. The latter consists of a fire-brick wall, 9 inches wide, and standing about 9 to 12 inches above the level of the grate-bars, with the upper corners rounded off. This forms a barrier to prevent the unburnt fuel from passing beyond the end of the grate, and by introducing a judicious degree of restriction in the area, the several layers of gases in the current are brought into contact with each other, thereby promoting combination, and effecting a better distribution of the gases against the heating surfaces. The exact height and form of the bridge are best determined by simple experiment. The height should not be such as to lead to any interference with the required strength of draft, nor so as to throw the hot current against the plates with such force as to show the least tendency towards over-heating; and it will naturally vary according to the consumption of fuel in the furnace. Usually the area over the bridge should not be less than three-quarters of an inch per pound of coal consumed per hour, and it need not exceed one inch per pound of coal. The bridge should never be built up of brickwork from the bottom of the furnace-tube, as this tends to cause corrosion of the plates against the lower part of such brickwork, by access of damp. The cast-iron bridge-bearer should be as compact as possible, but so as to close the opening quite tightly. The whole should be arranged so that as much of the plate as possible may be seen, and the whole kept free from accumulations of dirt. Many arrangements have been made for the admission of air at the bridge, with a view to improved combustion and the prevention of smoke, but these have on the whole been only moderately successful, and it is fully

established that all necessary air may be admitted through the fire-bars and at the furnace-door. If, however, any openings for this purpose are provided at the bridge, they should be lined or protected by cast-iron backed with fire-brick. By this arrangement smoother surfaces, superior curves, more accurate proportions, and freedom from change, by wear and damage, are secured. A soot-door is sometimes provided in the lower bridge. This is a convenience, but not a necessity, and is apt to lead to excessive admission of air. Special arrangements of fire-bars are described in the chapter on fuel and combustion. The furnace-fronts are secured by studs and bolts to the boiler-front, and provided with doors to open for stoking. These doors should be provided with openings of an aggregate area of about one square inch per 8 or 12 pounds of coal burnt per hour. These, being permanently open during work, admit air above the fire in quantity sufficient to prevent any serious amount of smoke production. Very soft, smoky, bituminous coal will require a greater amount of air admission at the door, while hard anthracitic coal, which allows the passage of more air through the fire, and which, moreover, is not so liable to serious smoke production, may be safely allowed less. In all cases, the furnace fittings should be so arranged that all air which enters here is directed on to the fire. If allowed to enter without control, the cold air is quite likely to pass along the top of the furnace-tube, so as to escape intermixture and combination with the gases from the fire. Thus perfect combustion is prevented, smoke is produced, and the plates are exposed to irregular heating, to the structural prejudice of the boiler. Martin's pattern of furnace-door, which is elsewhere referred to, gives a very good curve along its face for the guidance of the admitted air. This door is

solid, it opens inwards over the fire, and it is apt to burn off at the end, when it may be left to admit air beyond the power of interference of the stoker. Perhaps this burning off of the door gives the best possible adjustment of the air admission for any particular case. A loose plate not hinged, well fitted, almost air-tight, should be employed to close all openings in the furnace-front, above and below, when work is suspended. The fire-bars are usually set with their upper surfaces at about the centre plane of the furnace, and quite parallel thereto. Sometimes they are set much lower at the back end, with a view to the provision of a sufficient height of bridge above the grate, without unduly trespassing upon the area over the bridge. Also to facilitate the working backward of the coal on the fire, the fresh coal being chiefly placed upon the front end of the fire. The latter reason possesses more force when mechanical stokers are used, but in any case, a fall of three or four inches from front to back may be wisely adopted.

Dampers.—Dampers are adopted for the adjustment of the draft, and thereby the amount of air supplied to the furnaces, the quantity of fuel consumed, and the amount of water evaporated. These should be simply arranged, to be worked from the stoke-hole. Either one or two may be required for each boiler, according to the type of boiler setting adopted. They may be arranged to slide vertically, and connected by a chain to a balance-weight in the stoke-hole, or arranged to swivel around a vertical spindle or shaft, and worked by rod and lever. The former is generally to be preferred, though the latter is convenient in exceptional cases. In the latter case the proportions adopted should be such as to allow access for a full-grown man past the damper to the chimney bottom. Therefore there should not be less than 11 inches width of opening

available at any point, to secure which it is sometimes necessary to place the spindle towards one side. Dampers which are likely to be exposed to great heat may occasionally require to be straightened, and should be so arranged that they may be withdrawn at any time without disturbance of brickwork. Steam dampers are actuated by the pressure existing in the main steam-pipe. When the pressure falls, the damper is opened, and conversely. Only one such damper is used for a range of boilers, all the tributary dampers upon the boilers, in work at any one time, being thrown wide open. By this means the amount of cold air drawn in about the economizer and the neighbouring flues is minimized, to the great advantage of the work. In the absence of a steam damper, the ordinary damper upon the main flue should be alone used for regulating the draft, for the same reason.

Pressure test.—A finished boiler is generally tested by hydraulic pressure to about 100 pounds per square inch above the intended working pressure, all fittings being in position except safety-valves, which are separately tested. The furnace-tubes are gauged while under pressure as to maintenance of circular form. The end-plates are also gauged for deflection under pressure, and the whole boiler examined for leakage or evidence of weakness. Water at a temperature of 150° to 180° F. may be used with advantage in the hydraulic test. An excellent practice is to stamp the front plate with date and particulars of test.

Stationary shell boilers of exceptional types.—The foregoing description applies to the majority of boilers provided during recent years in good English practice, and as shown in Figs. 9, 10, and 11. But under exceptional conditions other types may be introduced. On the Continent the elephant boiler is

largely adopted, in which one large steam vessel is connected with two or three smaller ones placed beneath the first, and containing only water. The whole is heated by a fire placed beneath one end, the gases from which are passed successively over every part of the surface of the compound boiler, including the steam space. The latter condition tends towards superheating of the steam, though it probably never in practice reaches beyond a slight drying action. The application of heat to the upper part of Lancashire boilers has been shown to be attended by serious accidents. Possibly the absence of accidents shown to be thus caused to elephant boilers may be due to the smaller diameter of shell, and consequently increased elasticity. The proportion of heating surface to grate surface in elephant boilers is about 25 or 30 to 1, which is about the same as in Lancashire boilers. The evaporative efficiency is about the same, and elephant boilers possess little or no advantage over Lancashire boilers except in the smaller width occupied. The ordinary return-tube marine, or Scotch boiler, is largely preferred for stationary use by those who are familiar with it on shipboard. The ratio of heating surface to grate surface is about the same as above, but owing to the fine subdivision of the gases among the tubes, this boiler evaporates rather more water per pound of coal than does a Lancashire boiler without economizer, but much less than a Lancashire boiler with economizer. A marine boiler occupies less floor space, but a greater vertical height than a Lancashire boiler of equal power. The greater diameter of the marine boiler prevents its transit by rail, and in some cases gives trouble in placing in position. It is indifferently adapted for dealing with dirty water, but no part of the interior is absolutely inaccessible for cleaning. The tubes of marine boilers should be spaced

sufficiently far apart to allow free circulation of water; this is of importance at all times, but especially so when the water is not good. Defective circulation leads to blocking of the spaces, over-heating, and chronic leakage around tubes in tube-plates. Boilers of the locomotive type produce large quantities of steam on a comparatively small floor space, and offer little difficulty in transport. The heating surface is from 60 to 75 times greater than the grate area, so that they give a high evaporative efficiency when moderately fired. By the application of blast, 80 to 100 pounds of coal may be consumed per square foot of grate surface; the total amount of water evaporated is increased, but the efficiency falls off as the quantity of heat absorbed per square foot of heating surface rises. The tubes are set more closely than those in a boiler of marine type, so that locomotive boilers are not adapted for dealing successfully with any water except of a high degree of purity. The interior of such a boiler is inaccessible except by the withdrawal of tubes.

Boilers of both marine and locomotive types are well adapted for working in connection with an iron chimney. The boilers of portable engines are usually small. Their heating surface is about fifty times greater than the grate surface, and such boilers may be regarded as a variety of locomotive boilers. In all locomotive boilers the grate surface is below the barrel of the boiler, and a good circulation of water is easily secured. Cylindrical boilers externally fired, and provided with a large number of tubes through which the gases are returned to the front and afterwards led along the sides to the chimney, may give large heating surface, and are efficient for use with good water. Failing this condition they are subject to great wear and tear in the parts immediately exposed to the fire. This arises from the excessive

accumulation of solid matter at the place where it is in a position to prove most injurious. To work such boilers to advantage it is essential that special attention be devoted to thorough cleaning of the boiler, especially with a view to prevent the dropping of flakes of scale from the tubes. Egg-ended cylindrical boilers externally fired are obsolete in ordinary work on account of their low efficiency and great wear and tear.

Sectional boilers. — In some situations it is quite impracticable to provide access to enable a boiler of any of the types already described to be placed in position. The adoption of a boiler of a sectional type is then necessary. In other cases such a condition may not be imperative, but may possess sufficient importance to call for observance. Many boilers of sectional types have been successfully adopted in cases in which the work is not too severe and is fairly uniform, and in which a supply of pure water is available. The several fittings necessary for sectional boilers are practically identical with those already described. Many types of sectional boilers were described by Mr. Milton in a paper read before the Institution of Naval Architects in 1893, and published in the several technical journals at the time.

CHAPTER XIII.

BOILER-HOUSES AND BOILER SETTING.

Doorways and posts.—In arranging a boiler-house, the doorways should be so disposed as to allow the removal of any boiler in the series, without disturbing any other one or removing any permanent stone or brickwork. Such provision will be the certain means of ultimate saving in cost and inconvenience. In many cases this condition is most conveniently fulfilled by the provision of one large doorway, with a continuous lintel supported by columns which act as door guides, and each of which is arranged for temporary removal to allow the passage of a boiler in or out. The removal of a column must be conducted with reference to the strength of the lintel and the load imposed upon it. All necessary columns within the building may be treated in the same way. Columns interposed between the brickwork of the several boilers should only be adopted in cases of absolute necessity, as they are apt to lead to leaky brickwork and to interfere with changes which may at some future time become desirable.

Levels and drainage.—The levels of the boilers and accessories are of great importance. Some regard will be paid to the surface level of the adjacent ground, but still more to the level of the water which may

exist in the ground, either permanently or occasionally. The levels of the drains available for service must also be regarded. The discharge-pipes from boilers should be sufficiently high to discharge freely, when washing out after cleaning; also those of the economizers. If this condition cannot be complied with, pumping must be resorted to.

Concrete bed.—In all cases, a continuous bed of close water-tight cement concrete should be laid over the entire area below the bottom flues, and the brickwork built upon this. Such a condition is imperatively necessary when any possibility exists that the ground may at any time develop dampness. Concrete work generally is treated in the chapter on foundations.

Floor surfaces.—A good floor should be provided, sufficiently smooth for easy shovelling, cleaning, and wheeling over; hard and tough enough to resist rough wear, and the dropping of grate-bars, &c.; and not too slippery for safe walking. Soft cast-iron is very good, presents a good appearance, and may be roughed sufficiently to give a good foot-hold. Cast-iron plates are most easily roughed by means of shallow grooves cut in the wood pattern and reproduced in the castings. Such grooves should, however, be made much wider than usual, or the surface is little—if any—better than a plain one. For the passage of horses, and for the greatest degree of freedom from slipping, the irregularity is obtained by bits raised above the surface, as in Fig. 13, and as generally adopted in weigh-bridges. If required for wheeling barrows over, these bits should be made much smaller than usual, and generally the several dimensions may be varied to suit the conditions of each case. Plates of wrought-iron are roughed by grooves turned in the rolls, producing ridges above the surface. They give a surface less slippery than

that of ordinary cast-iron plates, but are more subject to corrosion, and in usual thicknesses are apt to

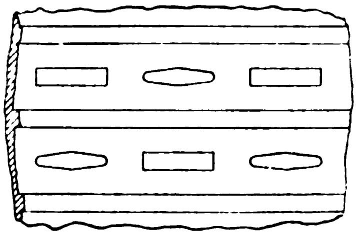

Fig. 13.—Unpierced floor-plate.

become twisted in wear. Open gratings are unsuitable for use as boiler-house floor-plates, but are useful for elevated gangways, and for engine-room floor-plates. In the plainest form these are made of wrought-iron, a series

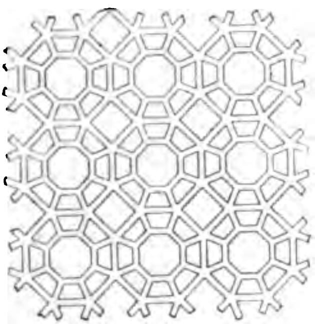

Fig. 14.—Pierced floor-plate.

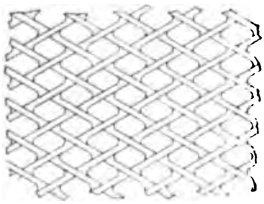

Fig. 15.—Pierced floor-plate.

of parallel round bars $\frac{3}{4}''$ or $\frac{7}{8}''$ in diameter being riveted into a flat bar at each end, the strength of the latter depending upon the size and loading of the piece. Cast-iron gratings are less slippery than those of wrought-iron, are less flexible, and present a better appearance. Two of the very large number of patterns adopted are shown in Figs. 14 and 15. In all cases in

which cast-iron is adopted for a floor or gangway, great care should be exercised in the design, with a view to minimize the number of sizes of plates, and to avoid irregularities such as those necessitated by the passage of pipes; otherwise very serious cost and delay will be involved. Each separate grate or plate which is placed upon bearers should be secured against slipping and falling when any or all adjoining plates are removed. Good cement concrete and hard stone flags are suitable for boiler-house floors when well laid to moderate slopes; but both are very troublesome when they prove soft. Brick is occasionally used, but difficulty is sometimes experienced in procuring bricks which are sufficiently hard and yet not slippery. They should always be laid upon a bed of concrete, or the surface will probably become so uneven as to cause difficulty in shovelling.

Floor beneath boiler front.—When Lancashire boilers are adopted of diameter not exceeding 7 feet 6 inches, the lower edge of the front-plate may stand just above the floor level. But if boilers of larger diameter are set in this manner, the fire-doors are too high for convenient stoking. It is at all times most undesirable that the front-plate should be covered by the floor-plate, on account of the corrosion which thus arises. A dust-tight joint is difficult to secure, and the presence of ashes in the blow-off pit is very objectionable. The difficulty is best met by sinking the floor in the form of a wrought or cast-iron channel along the front of the boilers, the width of which should just suffice for sweeping clean. The adoption of such a channel practically prevents the raking of the ashes from the fires on to the floor, but this is an advantage rather than otherwise, as in good practice the ashes are always raked into a barrow or light railway truck and carried away at once.

Light and ventilation.—Good light and air should be

provided in a boiler-house, to promote the comfort of the men and to attract the best men available. An opposite condition tends to cultivate neglect in every possible form. The front of the house will probably be entirely occupied by doors, but these may be hung on chains to open freely after a short lift; and they may be fitted with windows of strong glass. In the absence of occupied rooms above the boiler-house, skylights should be provided to give light on the tops of the boilers, but the boiler fronts may be well lighted by glass in the doors, which is at all times efficient for the purpose, and is less liable than skylights to be neglected in cleaning. Ventilation may be provided in the roof, over the doors, at the eaves, or in the several walls. The minimum area to be provided is about the same as the area of chimney opening, or 1½ square inches of opening per pound of coal burnt per hour. The total amount of opening should be divided so that different portions may be exposed to different winds, when the fires will have a chance to burn equally in all weathers. If the openings cannot be divided equally in all directions, a special effort should be made to avoid directing the principal ones away from the direction of the heaviest winds. All openings should be arranged to close when work is stopped, and thus prevent unnecessary loss of heat.

Brick walls.—The walls of a boiler-house should be of brick, laid in good mortar or cement. Doors, sills, and steps should be of hard stone. All angles, door-quoins, and other parts exposed to rough usage or contact with vehicles should be built in the hardest brindle or blue brick, or protected by stout cast-iron blocks or plates. The boiler settings should be kept separate from the side-walls of the house—and better if kept separate from each other—owing to the expansion of

heated brickwork. This separation is all the better if made by means of an air-space. Authorities vary much as to the expansion of brickwork, but experience shows that it should not be taken as less than $\frac{1}{1000}$ part of the length for a rise in temperature from the freezing-point of water to its boiling-point, or a difference of 180° F., while that of cement concrete is probably twice as great. Cavities in the walls of the building weaken them, to no possible purpose. All walls which are exposed to racking by expansion caused by changes in temperature should be kept up to strength, so that openings for windows or other purposes should be made as small as practicable, and so situated as to avoid the parts most exposed to such action.

Buildings of low cost.—Buildings of galvanized iron or other light construction may be adopted with a view to reduction in the original cost. These, however, cause the dissipation of much heat, they cost much more in maintenance, and they very soon present an appearance of dilapidation. In good practice, their use is only admissible as a temporary measure. In any case, the use of wood posts and framing should be avoided, on account of fire risk.

Roof.—The roof of a boiler-house should be of iron (or steel) construction, boarded and covered with slates or tiles. A roof of wooden construction is objectionable on account of inflammability, and of the mass of dust which collects on all surfaces which are sufficiently flat to hold it. The slates and boards are sometimes spaced apart, so as to give a certain amount of ventilation. This, however, cannot be adjusted, and so is apt to cause much loss of heat during the night, when the draft is stopped. A covering of galvanized iron costs less than slates or boards, but leads to a greater loss of heat in the roof than when adopted for the sides of the building.

Spare boiler.—Unless prevented by the most imperative necessity, provision should be made for a spare boiler, so that there may be no excuse for blowing-off a boiler before cooling. While a spare boiler gives special facility for the deliberate, workmanlike, and economical execution of all necessary cleaning and repairs, it has an equally important effect in the reduction of the total annual amount of repairs. Some regard should also be paid to the possibility of future extension.

Plant for purification of water.—When hard water is used for boiler supply, there is a great advantage to be secured in the adoption of plant for water purification. In the majority of cases there may be a disinclination to adopt such plant in the first instance, or to incur any expenditure in anticipation of such adoption. Though the space occupied by such plant is not large, it is well, in the design, to keep in view the desirability of its ultimate provision.

Concentration of drainage from cocks.—Usually there is a considerable amount of open space adjacent to a boiler-house. In such cases, an endeavour should be made to lead all waste-pipes from the boilers and engines to one common point, where they may deliver into a small open tank, at a convenient level. It thus becomes possible to see whether all cocks and fittings are tight. An inspection of the discharge from the pipes leading from the blow-off cocks and scum-cocks will be of assistance in forming an opinion as to the length of time during which blowing should be continued on each occasion. The water discharged from most of the pipes may be again used, but that discharged from the boiler blow-off and scum-pipes should always be sent to waste, except from absolute necessity, or when a water-purifying apparatus is employed.

Boiler setting.—The great majority of boilers of

Lancashire or Cornish type are set in the same manner upon two walls as shown in Figs. 9, 10, and 11, built longitudinally along the bottom, and at a distance apart of nearly half the diameter of the boiler, the clear height of this passage being usually rather less than its width, and the sectional area approximately equal to the combined area of the two furnace-tubes, or slightly greater. Upon the side-walls, the boiler is carried by the intervention of fire-brick blocks, shaped to give a bearing of about nine inches in width upon the walls, and a bearing of about four inches in width against the boiler. A bearing of this width is found to be amply sufficient for all purposes, and yet to give good access for the heated gases to the boiler shell. There is also less harbour for damp to accumulate against the boiler shell and cause wasting of the plates than is found to be the case with a wider bearing. There is a large proportion of the total surface of the boiler which is conveniently accessible for the necessary periodical inspection of the plates. As an additional facility in this respect, the bearing-blocks in way of the ring-seams of the boiler are often partially cut across before burning, in such a manner that after they are fitted in position beneath the boiler a small piece may be broken off each. The removal of these pieces at each time of inspection allows complete sight of every part of each ring-seam throughout the lower part of the boiler. The weak point of this facility is that the replacement of the bits may be occasionally overlooked, when some amount of heat will be wasted by the gases making a short circuit. With a view to reduce such possible loss, it is well to make each opening very small, especially at the inner edge. The bottom walls are arranged to carry the two lines of bearing-blocks quite to the back end of the boiler; and at the front end only

leave a sufficient space for communication with the side-flues, which are formed by means of side-walls built vertically at a distance of from 10 to 12 inches from the side of the boiler. The latter width is found generally suitable, as on the one hand it confines the current of heated gases, so that they make effective contact with the boiler, and on the other hand gives good access for examination of the boiler to the top of the side-flues, while with large boilers the sectional area of flue is not excessive. The floors of the side-flues are almost level with the bottom of the boiler. The tops of the side-flues are formed of arched fire-brick blocks set with the underside a few inches below water-level in the boiler, so that the heated gases do not come into contact with any part of the steam space of the boiler.

Alternative arrangements of flues.—The gases are generally passed from both internal flues, at the back end of the boiler, downwards into the one bottom flue and brought to the front end in one stream. At the front end they are separated into two portions, one of which passes along each side-flue to the back end, where they may be combined into one. Usually, however—and especially where several boilers are worked in connection with one chimney—the two side-flues are separately connected to the main flue, and each has a separate damper. Sometimes the gases are directed along the opposite course; they are divided at the back end, on leaving the internal flues, and one portion passed along each side-flue; at the front end they are re-combined and passed along the bottom flue to the back, where one damper is used to control the whole. Throughout the entire circuit from the fire to the point where the gases leave the boiler they are in contact with the heating surfaces of the boiler, to which they are imparting heat. This causes a continuous fall in

the temperature of the gases. Therefore under the first arrangement, in which the gases pass beneath the boiler directly from the internal flues, the bottom of the boiler is exposed to a higher temperature than under the second arrangement, where they reach this part only after a passage along the side-flues. Under the first or ordinary arrangement, therefore, the boiler is exposed to the hottest gases beneath, to gases of reduced temperature at the sides, and at the top only to the temperature corresponding to the working steam-pressure. Under the second arrangement the boiler is exposed to the highest temperature at the sides, to a reduced temperature beneath, and at the top to the same temperature as under the first arrangement. The boiler shell is therefore more severely strained by expansion under the second arrangement than under the first. But the bulk of solid material which enters a boiler, from whatever source, tends to settle to the bottom. When great heat is applied to the bottom of a boiler, beneath a deposit of solid matter, a risk of over-heating is at once incurred. Under this condition, the existence of even a soft impalpable deposit upon the plates is objectionable, but if a piece of scale, or any other solid body of appreciable size, should have been overlooked and left inside the boiler, serious damage may arise. A piece of waste possessing low conductivity for heat is capable of causing more mischief than solid matter of close texture. Experience shows that externally-fired boilers give great trouble in this way, but in these the boiler shell is exposed to much higher temperatures than the shell of an internally-fired boiler. The ordinary arrangement also promotes good circulation. A due consideration of all the conditions, in combination with the teachings of experience, will generally show that for long boilers, in which the furnace

gases have much heat extracted before leaving the internal flues, and in which there is the greater necessity for paying attention to the effects of expansion by heat, it will be wise to adopt the usual arrangement. In either case, when raising steam from cold water, the bottom of the boiler rises in temperature at a much earlier period than in some other types, though still leaving very much to desire in this respect.

Construction of cavity behind boilers.—Whichever course is adopted as to the direction of the draft, the construction of the flues is precisely the same throughout, except at the part which is behind the boiler. In either case, the roof over the back cavity is the most difficult part to make quite secure. Different constructions are adopted, including arches and flat tiles supported by over-sailing brickwork at the back, and by bearers against the boiler. Probably the best arrangement is to reduce the opening to little more than half by over-sailing, and to use a heavy angle-bar with the vertical table standing above, so as to avoid exposure to greatest heat; a channel bar may also be used in the same way. Either will give a good support for the flat tile-covers. The roof should be sufficiently strong, without the necessity for support from the intermediate walls; but these should be tightly fitted against the roof, so as to prevent leakage of gas. A vertical wall is sometimes built in the space dividing the gases from the two furnaces until they reach the bottom flue. This steadies the draft and gives additional security to the roof.

Disposition and structure of walls.—In all cases, the thickness of brickwork in contact with a boiler must be as small as possible; very seldom indeed does it require to exceed 4½ inches, even in the front vertical wall: thus, no part of the boiler, inside or outside, except the

top, should be covered out of sight over a greater width than 4½ inches, so that any point is within 2¼ inches of sight. The front wall must be built to avoid contact with the angle-ring which attaches the front plate, and continued beneath the boiler in such a manner as to avoid covering the blow-off cock, elbow-pipe, or mounting-block. These must be left so that their rivets, bolts, and all parts are visible and free from restraint by brick-work, when expanding or contracting, as such restraint would almost certainly cause breakage. All brickwork in contact with flue gases must be of fire-brick, set in fire-clay. In work of great strength exposed to little heat, and in the level foundation, Portland cement may be used, but never any kind of lime mortar, in proximity to the boiler, as it disintegrates under heat, and it is apt to collect damp, and thus promote the corrosive wasting of the plates. Common stock-bricks may be used for backing the walls, but they require to be of the same thickness as the fire-brick facing-bricks, so as to bond together. Usually there is very little saving in the use of two kinds of bricks, unless the fire-bricks are of an unnecessarily costly description. The facing-bricks should be bonded by headers about every fourth course. These give greater security in work, and facility in making repairs. Glazed bricks may be used for the front wall. These may be of any colour. White give the best appearance while fresh, and if well laid and jointed, but they soon deteriorate in wear.

Necessity for access.—Throughout the work, full consideration must be paid to the necessity for proper access to every part of the boiler and flues, sufficient for a man of ordinary build. This is most necessary, as it is not always possible to arrange for a man of less than average size to undertake the work of cleaning, and the only safe course is for occasional examinations to be

made by some one in a position of responsibility. The knowledge that this may be done at any time, and upon any part of a boiler—inside or outside—is a strong incentive towards the maintenance of all parts in clean working condition. In the absence of this, the attendants are very apt to under-estimate the necessity for complete cleanliness, and the chief engineer or manager who takes the trouble to make a personal examination generally considers the time to be well expended. Further advantages in connection with attention to minute leakages, &c., are obvious. No opening should therefore be less than 15×11 inches, with convenient approach or means of support at each side.

Development and consequences of leakage of air.—Brickwork, flooring, and all other parts should be tight so as to prevent all indrafts of cold air, to the detriment of the chimney draft. This is most likely to occur at the doors beneath the boiler front for access to the flues, or at the slits through which the dampers work, or at the chain-holes in the economizer; but it may happen at any point in the brickwork, either as an original defect, or one developed by subsidence due to defective foundations, or by changes in the condition of the brickwork due to changes in temperature. The simple expansion and contraction of brickwork has been already referred to. But brickwork does not always contract upon cooling to the same extent as it expands by the application of heat. This is sometimes observed as a result of atmospheric changes in temperature, in coping courses laid upon brick walls, but it is found to be very much more strongly developed in brickwork exposed to high temperatures. Some of this action is probably due to molecular changes in the substance of the wall, but a large proportion is in many cases due to

a disruption of joints and entry of sand or other material which cannot be removed, and which causes the total length of brickwork to reach increased amounts on each successive occasion.

Passages, curves, and surfaces.—All flue surfaces should be finished to a good smooth and even face, with curves well faired to each other and of ample radius. Dead-ends should be avoided. The sectional area of flue should be as uniform as possible, so that the gases will at every point move as nearly as possible at one uniform velocity. By this means the draft will be improved, and irregular deposits of dust will be avoided or reduced.

Special setting.—Special systems of setting are sometimes adopted. In one system, the boiler is set on iron stools, placed beneath the centre line of the boiler. The draft is arranged to run down one side of the boiler, and up the other one, and the sectional area of the flues is arranged to increase with practical uniformity throughout the circuit. The general efficiency of a boiler set under this system is found to be very good, but no advantages appear to have been secured.

Necessity for dryness.—A boiler should never be placed in a damp situation, except from the most absolute necessity. Any serious degree of dampness militates against the durability of the boiler, and against its efficiency as a steam generator, as some of the heat developed in the furnace is wasted in evaporating water which ought not to be present. The chimney draft also becomes impaired in consequence of the presence of water-vapour, and the prevalence of a lower temperature than would otherwise obtain. Obviously this trouble will be found to vary in different cases, but in all cases dampness is objectionable. Sometimes water is found to stand a few inches deep in the flues. When such

water is drained away, a very distinct reduction is made in the coal consumption.

Adjustment and temporary support in position.—A boiler should be placed in position with the greatest accuracy before the building of any part of the walls is commenced; otherwise, no amount of care can prevent serious damage to the walls, or the bedding of the seating-blocks. The floor of the bottom flue is of less importance, as any damage which it may suffer may be afterwards repaired. The boiler being placed in position, with the correct amount of fall towards the front end (probably 1½ inches), requires to be well supported for the building up of the brickwork and seating-blocks, which must be fitted just tightly in position with a thin bedding and facing of fire-clay. The best temporary support of the boiler is one of well-bedded wood blocks, crossed and re-crossed, so as to make the whole rigid in all directions. The top block of each set should be cut hollow to fit the curve of the boiler, and one set of packings should be placed under the boiler near each end. The bottom packing-blocks in each set should be bedded upon a thickness of well-rammed sand. When the work has fully set, the sand may be raked out so as to allow the boiler to take a bearing all over, absolutely without shock or appreciable movement.

Warming and covering of boilers.—When the whole of the brickwork about a boiler has been completed, and allowed to dry as thoroughly as the conditions will admit of, it may be filled with water and fired very slowly. This process is protracted as long as possible, after which it is fired up to make steam. After some days'—or possibly weeks'—working, the top of the boiler is covered with one of the many compositions for the retention of heat, which is also applied to the steam-pipes. The front-plate is very seldom covered in any

way, on account of the numerous fittings distributed over it, and because most coverings would be likely to conceal any accidental leakage which may arise. Removable block coverings may however be used with advantage.

Tests for leakage.—Leaky brickwork is sometimes revealed by the production of dense smoke in the furnace, and stopping the damper. This test is, however, of little use in showing leaks at a lower level than the furnace; but when it does show one, it indicates the precise locality. In all cases, analyses should be made upon gases collected at a point where they could not be subjected to dilution by reason of leakage, and at a second point near to the foot of the chimney. A good point for the first purpose is found beneath the boiler. A small hole may be made in the brickwork in the ash-pit and a long pipe inserted. The hole in the brickwork around the pipe should be carefully stopped, to prevent inward leakage of air. Tests upon the tightness of boiler brickwork should be carefully performed, soon after completion of new work or repairs, and at intervals afterwards.

Qualities of fire-bricks employed in boiler work.—The seating-blocks and brickwork in connection with internally-fired boilers are exposed to such a temperature as to require the facing—at least—to be of fire-brick, set in fire-clay. The fire-bricks need not, however, be of the most refractory kinds, which are unnecessarily costly. These are only produced in few districts, while bricks of quality sufficiently good for the purpose are produced in almost every colliery district from the fire-clay or under-clay lying beneath the coal-seams. The majority of such bricks are, however, not sufficiently good for the construction of furnace-bridges or the entire construction of furnaces for externally-fired

and sectional boilers, or the arches, chambers, &c., usually necessary in firing with liquid or gaseous fuels. These should be constructed of Dinas, Ewell, Sheffield, Stourbridge, Newcastle, or equivalent bricks. The Ewell bricks are especially good for arches and similar work, by reason of the facility with which they may be rubbed to an exceedingly fine joint, so as to secure a mass practically homogeneous in character. Only the first quality of Stourbridge bricks are adapted to the work in question. Other qualities are useless on account of the admixture of common clay and other material. In the chapter on gas firing, it is explained that fire-bricks in furnace work suffer mechanical injury from contact with the stoking tools, chemical waste from a fluxing action in the ash, and physical destruction from cracks arising from variations in temperature. A careful examination will generally suffice to show how any injury has arisen, and to suggest measures for preventing any recurrence.

CHAPTER XIV.

CHIMNEYS FOR BOILER SERVICE.

Height of chimney to suffice for prevention of directly offensive discharge of gases.—The chimney attached to a range of boilers is required to act as a conduit, to carry away the products of combustion to such a height in the atmosphere as to avoid causing any annoyance to the public. With this object, most municipal authorities prescribe a minimum height for such chimneys, which is usually less than that rendered necessary by the second condition. Many years ago, when smoke and chemical fumes were discharged into the air with greater freedom than at present, chimneys were constructed of enormous height (300 to 450 feet) as a temporizing measure. But the Acts of Parliament and local bye-laws which now apply to this subject give no credit for exceptional height of chimney. In all other respects a height of 180 to 220 feet is usually found to suffice, and is therefore seldom exceeded.

Height of chimney to ensure efficient draft.—The second condition is that the chimney shall so direct the gases contained within it as to give rise to a current or draft of sufficient strength to supply fresh air to the furnaces as rapidly as it is consumed in the combustion of fuel. This is effected by reason of the difference in

density between the air outside and the gases within the chimney. In good practice the temperature of the gases inside a chimney is about 350° F., and the density of the mixed gases at this temperature is about two-thirds that of air at ordinary temperatures. The difference in the pressure is therefore equal to one-third of the weight of a column of air of the height of the chimney. The weight of air is ·081 pound per cubic foot. The gross pressure of air due to a column in a chimney 180 feet high is ·101 pound per square inch, or equal to a column of water 2·80 inches in height. That of the gases in the chimney is two-thirds of this, and the difference would be measured by a water column of 0·93 inch, if the gases were at rest. The velocity of air flowing through an opening into such a chimney, and meeting with no resistance in addition to that caused by the gases, would be the same as that acquired by a body falling freely through a distance equal to the difference in the height of column, or in this case 60 feet, which is the working head. The velocity thus acquired is 62 feet per second, or 42 miles per hour.

Area of chimney opening.—One pound of coal requires for its complete combustion about 12 pounds of air. If the gases produced in this combustion could be propelled up the chimney at the velocity stated, the minimum cross-sectional area of the chimney opening would require to be ·15 square inch per pound of coal consumed per hour. But the resistance to the motion of the gases which is encountered in the grate-bars, the fuel in the furnace, the flues, and the chimney, reduces the actual velocity, so that the area of chimney opening which is required is for this reason about five-fold greater than under the ideal conditions first assumed, or making it ·75 square inch. But in practice it is found necessary to be prepared to supply about twice as much

air as would suffice to satisfy the chemical requirements of the fuel. This raises the chimney area to 1·5 square inches per pound of coal burnt per hour, which is found to agree with good current practice, for cases in which 20 pounds of coal are burnt per square foot of grate surface per hour.

Existing chimneys of deficient proportions.—Many chimneys are in existence which provide much less area than that just given. But they were built when the gases were sent up the chimney at a higher temperature than is now allowed, and they are quite unable to produce draft sufficient to deal with 20 pounds of coal per square foot.

Margin in area.—In deciding upon the dimensions for a new chimney, it is most desirable that they should be, beyond question, ample. Every improvement in the abstraction of heat from the furnace gases leaves them cooler, and thereby tends to impair the draft. But, on the other hand, any process which facilitates a reduction in the amount of air supplied to the furnaces acts by way of relief to the chimney. But such relief is of no value in this connection if by any reason it should fail to be continuously maintained. In connection with this question, the possibilities of future additions should be fully considered, and in case of doubt the inclination should be towards amplitude. Many chimneys are worked to great disadvantage and loss by reason of successive additions of boilers beyond those originally intended, and which load such chimneys to the verge of failure.

Exceptions allowable.—The foregoing figures refer to chimneys of 180 to 220 feet in height, which may be taken as good practice. In many cases, where the work is lighter, a chimney of less altitude suffices quite well. In other cases, in which the work is irregular, and full

power is only required for a short time, the inconveniences attendant upon a short chimney may be endured.

Height as affecting draft power.—The draft power increases in accordance with the square root of the height of a chimney, somewhat modified by the additional cooling which occurs in a very high shaft. If a siphon gauge is applied to the bottom of such a chimney as described, the draft should support a column of water of from five to seven-eighths of an inch. If this is not reached, there is probably a leakage of air taking place through the brickwork dampers, economizers, or elsewhere. Such a leakage of air will quench the draft by cooling the air in the chimney, and by increasing the amount of gases to be dealt with. A high-water gauge is the chief indication of good working, but a good draft as indicated in this manner may be wasted by reason of injudicious proportions in the flues, either by way of providing insufficient area at some point, or by the occurrence of large spaces where the gases may eddy or become stagnant, involving resistance in restarting on their course. Abrupt angles and rough surfaces have also a similar effect. If the boiler-house is not well arranged, the draft will be injured during a gale blowing from certain directions.

General construction.—The pedestal of a chimney is usually vertical, and the shaft built with the outside to one uniform batter, which in different cases is from 1 in 20 to 1 in 40, or 1 in 45. In all long chimneys the thickness of the wall is reduced upwards, either uniformly or in a series of steps or offsets, according to the construction. In all ordinary cases the steps are made in the inside, so that the area of opening is varied at each one. As a rule, however, the area is much greater at all other points than at the top. Very short chimneys

are sometimes built quite vertically on the outside, with the inside wider at the top than at the bottom.

Calculation upon weakest points.—Exhaustive calculations should be made as to the strength and safety of a chimney in all respects, in the foundation and the several parts of the superstructure. In a chimney of brickwork, the several steps at which the thickness of walls is reduced are unavoidably the weakest points in the structure. The calculations are therefore directed towards these points, and account must be taken of the action of weight and of wind pressure upon the whole.

Variation in point of greatest weakness.—In most cases, accidents arising to chimneys are caused by the overturning of the entire structure at or near the base. In other cases rupture takes place at a considerable distance above the ground; and occasionally near the top. As a rule, a chimney of ordinary construction will prove sufficiently stable if the width at ground level is about one-ninth part of the total height above ground. The width across walls at the top may then be made equal to one-twentieth part of the height. The average batter over pedestal, shaft, and cap will then be about 1 in 33. If the width across the base is less than one-eleventh part of the height, the chimney will almost certainly prove weak. The former proportions give a good, harmonious appearance. A shorter chimney is stronger, but presents an inferior appearance, and, unless designed and built with special care, is deficient in draft power.

Hollow batter.—A hollow batter, or one in which the upper part is more nearly vertical than the lower, is sometimes adopted with a view to spread the base and improve the foundations. This measure might be much more frequently adopted, with advantage to the security and appearance of the chimney.

Primary condition to be observed.—The area of opening at the top of a chimney is generally the smallest, and therefore must be chiefly regarded. The whole design may also be commenced at the top and worked downwards.

Form of cap.—The shape of the cap should be such that the draft of the chimney will never be baffled in a gale. With this object, the upper part of the cap should be formed to a concave shape, so that the wind, striking it horizontally, will be diverted upwards, and tend to increase rather than to retard the draft, thus virtually adding somewhat to the height. This shape of cap is very similar to that adopted by many engineers for the funnels of locomotive engines. Such a cap also passes the wind much more freely than one of most other designs, and thereby throws less stress upon the structure in a gale. Many chimneys are provided with caps of a convex form at the top, when it sometimes happens in a gale that the wind will follow the surface, and bury the lee side in smoke half-way to the ground. In such cases the draft unquestionably suffers. In a manufacturing district, instructive observations may be made upon the effect of each type in a gale. The effect is modified by the force of the escaping gases, and occasionally by proximity to high buildings, but the general balance will be found to be distinctly in favour of the concave type of cap.

Diaphragms in cap.—Cross-walls carried above the shell of a chimney, and in form resembling the cross-section of a bee-hive, are occasionally employed with a view to prevent the baffling of the draft in a gale, in which respect they are undoubtedly useful. But they cause an important contraction in the area of opening, they present additional area to the wind and so increase the overturning power of the wind, and they apply a

bursting force upon the top of the chimney. The last is the most serious objection, even though it may be more or less completely met by special means for tying the whole structure together. Iron plates for the same purpose are almost free from the first and third objection above referred to. But they are especially subject to corrosion, on account of their position.

Construction of cap.—A chimney-cap should be of moderate weight, to avoid the crippling stresses which a heavy one imposes upon an oscillating chimney in a gale, and which have been found to be the direct cause of accidents. The cap may be of sound sandstone, or of such limestone as is quite free from liability to crack by the action of frost. Better still, it may be made of special fire-brick block, or of terra-cotta. But well-proportioned cast-iron is best of all. Whatever material is adopted, the whole should be carefully rounded, with a view to avoid the formation of cracks. It is most important that the whole should be well secured together and to the brickwork. If made of cast-iron, the cap may be left hollow, and faced up with brickwork to give a flush surface to the bore of the chimney. Fig. 17 shows a section of such a cap for an octagonal chimney 9 feet wide outside brickwork. The cap is cast in eight pieces, bolted together at the angles, and also secured by a hoop of flat bar-iron, inside, near the top. Chipping-strips should be cast on each segment, and the whole fitted together on the ground, upon a template made from the finished brickwork; then taken apart, hoisted, and re-fitted; the whole to be bedded on cement mortar, the joints grouted with cement, and inside work paid over with neat cement wash after bolting together. Nine-inch brickwork to be then built inside the cap, and the top filled with fine cement concrete, to surround the hoop in a perfectly water-tight

manner, convenient holes being made in the castings for this purpose. In grouting the cap with cement, all leakage upon the outside should be carefully avoided, as cement wash is difficult to remove from brickwork, and ruins the appearance if allowed to remain. Glaziers' putty is sometimes very useful in stopping joints for work of this kind.

The cap just described, inclusive of brickwork, cement, and concrete, weighs only about the same as an equal height of 13½ inch brickwork, notwithstanding that an ample projection is provided. In any solid material it would also be very much lighter than many types of cap in use.

Upper brickwork.—The brickwork next below the cap should be built almost or quite plumb, and set in cement. The string-course should be of fitted blocks of hard stone, and should project a few inches for the sake of appearance. A similar course of stone next above the projecting string-course should break joint with it, should be set in cement, and stand flush with the brickwork.

Strength of brickwork in shaft.—Below the string-course is the plain, battered brickwork, in lengths, each thicker than the one above it by one half-brick. The several lengths should not exceed 25 to 40 feet each for chimneys of ordinary dimensions. In very tall and wide chimneys, the several lengths are increased with advantage. The calculations will show the strength of the chimney at each change in thickness, these being the weakest parts of the whole. The dimensions may be varied generally, in accordance with the results of such calculations. But in most cases this will be most readily adjusted by raising the step to a higher point when the chimney is weak, and, conversely, by lowering it when the strength is shown to be in excess. When

the positions of these are definitely decided upon, a strict observance of them in erection should be ensured. The area of each section must be such that the pressure upon the brickwork due to the mass above it shall not exceed a safe amount per square foot. Very good bricks, well laid in mortar made from lias lime and clean sand or substitute in the proportion of 1 to 2, will safely bear a pressure of 8 tons per square foot. Less perfect work will not safely bear more than 6 tons. In ordinary work the pressure upon brickwork is assumed to be uniformly distributed. In chimneys and similar structures this is also the case in ordinary weather. But in heavy gales the pressure of the wind applied to one side, acting in combination with the weight of the structure, greatly disturbs this condition. The total weight imposed upon any particular section remains unaffected, but the brickwork towards one side suffers an increase of pressure, while that on the opposite side is correspondingly relieved. In an extreme case the latter action may be continued so far that the brickwork is exposed to tension instead of compression. By its nature, brickwork is not well adapted for sustaining tension, and the occurrence of tension implies an increase in the total amount of compression applied to the section, simultaneously with a reduction in the area under compression; so that it is desirable to avoid the incidence of any tensile stress upon any part of the structure. It is found that this result is achieved when the centre of pressure resulting from the combined action of the weight and the wind-pressure upon the structure falls within a certain distance of the centre of figure. For figures geometrically similar, this distance bears a constant relation to the total width of the section, whether the same be of large or small dimensions. Fig. 16 shows this distance

graphically for square, octagonal, and circular chimneys of various proportions. The ratio which the thickness of wall bears to the total width is plotted in a horizontal direction, when the height of the proper curve directly over such point gives the maximum displacement of the centre of pressure from the centre of figure. This is found as a fraction, to be multiplied by the total width,

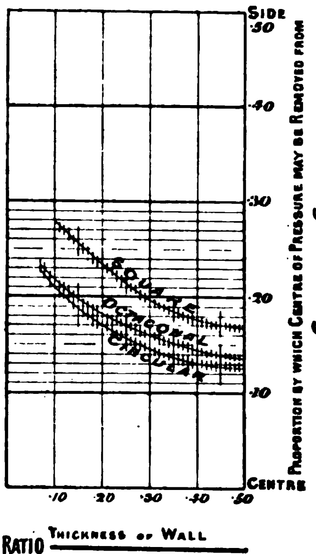

Fig. 16.—Maximum displacement of centre of pressure in chimney.

in order to find the distance in feet. Thus, if the thickness of walls is 20 per cent. of the whole width, the fractional distance is ·213 for a square chimney, ·182 for octagonal, and ·170 for a circular chimney. If in each case the total width is 12 feet, the actual measurements are 2·556, 2·184, and 2·040 feet respect-

ively. The calculation is described and illustrated in connection with an example appended to this chapter.

Comparison of sectional forms.—In the majority of cases chimney-shafts are square, octagonal, or circular. Hexagonal, star-shaped, and other forms have been used, but very seldom. A square shaft is the cheapest, and may be used for temporary or minor purposes, one advantage—sometimes of importance—consisting in the fact that no special bricks are required for the shaft. A circular shaft opposes the least resistance to the wind and to the current of gases in the interior, the latter advantage being, however, usually too small to be regarded. The stresses arising from wind-pressure upon square, octagonal, and circular chimneys of equal dimensions stand in the ratios 10 : 7 : 5. A circular chimney is properly built of special bricks throughout. For this reason, difficulties may arise as to the supply of bricks, unless adequate arrangements are made, well in advance of the work. There is, however, less risk of the bricks set apart for the purpose being accidentally sent to other work. An octagonal chimney requires special bricks for the angles, but all others are ordinary rectangular bricks. Octagonal and circular chimneys of equal dimensions are about equally costly. Opinions vary as to the appearance, but most persons prefer the octagonal form.

Brickwork generally.—Bricklayers employed upon the erection of a chimney should be well skilled, and fully understand and observe the necessity for good bonding. It is also equally important that the bricks should be well bedded and filled up solid. Special care should be exercised in the selection of the bricks, with a view to durability, and all soft bricks rejected, while the softest of those allowed should be used where they are neither exposed to the gases inside the chimney

nor the weather outside the chimney. Hard blue bricks are excellent for this work, provided that they possess a suitable surface for the adhesion of the mortar or cement.

Masonry.—Stone has an advantage over brickwork for a chimney, in the fact that the inside may be built flush or without break from top to bottom. A chimney built of ashlar stone throughout would be an ideal one. But stone chimneys are usually built with coursed face-work and rubble backing, which is very inferior to brickwork, in respect to homogeneity and strength. Stonework possesses an advantage in its greater weight per cubic foot, which imparts greater stability against wind-pressure. But in case a stone chimney should be reduced to the necessity for straightening operations, these are executed with greater difficulty than in a chimney of brickwork.

Changes in surface to be avoided.—In all chimneys an important condition to be observed is that panels or breaks of any kind in the continuity of the walling should be scrupulously avoided. The adoption of these opens the way for apparently slight negligences in workmanship, which may lead to serious consequences. Efficient bonding of panel-work is difficult to perform, and it is sometimes done in such a manner that the whole is left in a condition very little better than a collection of separate columns, the danger of which is obvious. Designs sometimes applied in glazed bricks are liable to similar weakness.

Mortar and cement.—Mortar made from highest quality lias lime and clean sharp sand, with a good admixture of coarse in it, is the substance generally used in best practice for setting the brick or stonework of chimneys, with the exception of the foundation and the cap, which are set in Portland cement mortar.

There is, however, every probability that the use of cement to the total exclusion of lime would be advantageous. But the use of the weak cement mortars which are sometimes so temptingly recommended cannot be too strongly discouraged. The mixture of mortar should be made in the proportion of two parts of sand to one of dry slaked lime, by volume. Regular samples of the mortar should be taken and allowed to set. The set samples should be broken and carefully examined for fragments of unmixed lime. These often occur where the inspection is not of the most rigid character, causing a mortar which is absolutely weak, though full proportions of lime may have been consumed.

Concrete.—Concrete has been successfully applied in the building of chimneys for withstanding ordinary temperatures. The colour and general appearance constitute probably the greatest objections to its use. Its use should not be attempted except in case of a circular chimney, owing to the trouble arising from fitting the moulds to suit external angles. If the staves are all made to one uniform width and taper, and used without change throughout the work, a hollow batter will be formed which in most cases will be preferable. The same drum-mould may be used from bottom to top for the bore of the chimney. The surface concrete should be finer than the hearting, and should be laid so that when the moulds are removed there are no cavities (large or small) left in the surface. This is promoted by keeping the concrete against the face rather higher than that in the heart during laying. If it is kept lower, a few stones incompletely covered with cement are apt to roll to the low parts, where they lie with small spaces beneath, which spaces are left hollow when the stones are

subsequently covered over. All set surfaces should be cleaned and hacked over before commencing to lay the next bed. Other precautions are referred to in another chapter, also in works specially devoted to the subject of concrete. No one should attempt to build a chimney in concrete without previous experience in concrete for other purposes.

There is every reason to believe that concrete will withstand a greater temperature than ordinary brickwork in mortar. But it should not be exposed to great temperatures in an important structure such as a chimney unless concrete of precisely the same composition has been previously tested by the application of heat. In any case, it is better to give a little more protection, by means of a lining of fire-brick, than is necessary for brickwork.

Concrete of good quality which is found to set well, but not too rapidly, may be loaded to a maximum pressure of 7 tons per square foot—corresponding in an accurately-proportioned section to a mean pressure of 3·5 tons.

Pedestal and connections.—In all cases the pedestals of chimneys should be of square form, with a view to sufficient strength after the deduction of the necessary openings. These openings should be made as narrow as possible, sufficient area being obtained by means of height. All arches and inverts should be made to a complete semi-circle, with a view to the reduction of the thrust to a minimum. Strong stone lintels may be used, but they are not necessary, and they possess some liability to crack on exposure to heat. In many cases it is found desirable to divide the flues, so that portions of the gases may be led into the chimney on two or three sides, the several openings being correspondingly smaller than one large one, and the local

heating of the shaft on one side being thereby avoided. In some cases a brick flue is carried completely around the base of the chimney, and three, or occasionally four, openings made into the latter. Sometimes such a flue is buried in the ground, the whole being supported on a large bed of concrete. This, however, should be of sufficient strength, having regard to the fact that such a flue weighs less than the earth which it displaces; therefore the chief weight is localized in the centre of a large plate, which may very naturally be made too thin for the work. A very good practice is to leave openings on all four sides of the pedestal, so that the strength of the structure is uniform all round. Some of these openings may be utilized in the first instance, and future alterations may be made without interfering with the strength or stability of the chimney. At least one opening should be made conveniently accessible for the cleaning and examination of the inside and the removal of dust. When the draft is admitted on two or more sides, diaphragm walls should be built, with a view to prevent eddyings, and one part overpowering another. These should be stopped short of the full inside width of the chimney, so as to leave access around the ends for cleaning. All openings of any kind in the outer walls should be carefully stopped, to prevent leakage of air into the chimney, which has a powerful effect in quenching the draft.

The square pedestal should be worked into an octagonal shaft by stepping. The best way is to effect this on the outside and inside, so as to preserve uniform thickness of wall throughout. If the outward appearance of this is considered to be objectionable, the pedestal may be finished square outside, provided the stepping is properly performed inside.

Buttresses against pedestal.—Where strength in the

pedestal is required, beyond that supplied by the square form, buttresses or pilasters may be built against the angles. These are occasionally desirable for the purpose of spreading the weight over a large area of foundation, on account of soft ground. Wherever these are adopted, they must be built in one continuous mass with the pedestal. If the square form of pedestal is considered to be objectionable, there is no necessity to carry it very far above the extrados of the arches over the openings.

Pressure upon foundation.—The foundation of a chimney must be adapted for supporting in safety a great pressure due to the weight of the structure acting vertically, and the force of the wind acting horizontally, the two combining to give an oblique action. The chimney may also become deflected slightly from the vertical condition, so as to still further remove the centre of pressure from the centre of figure of the foundation. Therefore the foundation must be so designed as to provide an ample margin of strength, and measures adopted appropriate to the different conditions which arise.

Rock foundations.—If a good bed of rock is found to exist within a moderate depth, and the superincumbent material is sufficiently dry and firm for excavation, the best way is to dig down to the rock and build directly upon it. Softness or weakness of the rock may be met by a proportionate increase in the area upon which the weight is distributed, the increase being obtained by spreading the footing courses of brickwork. For this purpose, a bed of hard consolidated gravel may be treated as a solid rock, according to its solidity and bearing power.

Pile foundations.—In case the hard bed should be covered by material which from its nature or attendant conditions may not be removable, piles of wood are

usually employed. These, however, should never be employed except when their tops are well below the level of permanent ground water. They would be quite safe if they could always be kept dry, but there is always sufficient moisture present, with a gentle heat from the chimney, to destroy the pile heads very soon, unless they are submerged in water. Good pitch-pine piles 12 inches square up to 35 feet in length may be driven so as to carry in safety 40 tons each. The last set per blow with a ram of one ton in weight, falling 8 feet, will probably be $\frac{5}{16}$ inches; but any decision as to safety will be largely influenced by the results of the preceding blows. All the piles should be driven and re-driven, as a precaution against lifting, before any are cut off. Then the water should be pumped out to 2 feet 6 inches or 3 feet below its natural level, the piles cut off level, fitted with cross-beams or head-trees, and the whole covered up with a bed of concrete upon which the superstructure may be built with security. The possibility that at some future time the level of the water in the ground may be lowered by improved means of drainage should be considered, and if necessary the piles cut off at a lower level, or the use of piles entirely abandoned.

Cylinder foundations.—In some cases cylinders may be used instead of piles. They may be carried to greater depths than piles of wood; and they are independent of the presence of water for preservation. An iron shell may be used, to be filled with concrete when the required depth has been cleared by excavation, either in the open or under air-pressure. A cylinder may also be made of concrete in sections, within which the excavation may be removed in the open. In some cases, where an iron shell is used, it may be withdrawn slightly in advance of the concrete,

as it is filled into the excavated hole. Cylinders of good ordinary cement concrete should not be loaded beyond 6 tons per square foot, though special material may be more heavily loaded. When cylinders are used, they must be calculated to sustain the entire load placed upon them, minus that of the earth displaced in the foundations. A chimney which failed in 1882 was built upon rough cylinders of lime concrete, which were loaded far beyond their strength, and which largely contributed to, if they did not actually cause, the catastrophe.

Soft earth foundations.—Where no bed of sufficient bearing power can be reached by piling or cylinders, the foundation must be placed upon the untreated earth, for which purpose an estimate must be formed as to the weight per square foot which may be safely applied. A bed of concrete or masonry is constructed of area sufficiently large to carry the total weight, and of thickness sufficiently great to impart transverse strength. The actual bearing power of the ground is much affected by the surroundings, and by the presence and fluctuation of ground water. Excellent hard, dry clay in level ground 12 feet below the surface may be loaded to 2 tons per square foot. Soft clay under the same conditions will not carry more than 1 ton. If clay is found to be hard when opened out, the possibility of subsequent access of water and consequent softening must be fully considered. A load per square foot which is always safe = weight per cubic foot of earth $\times$ depth from surface in feet $\times \cot\left(45° - \frac{\text{natural slope}}{2}\right)$. This load may be always exceeded, but it is impossible to give any exact rule as to the amount, except that in very bad foundations the possible excess is so small that it should be neglected. Gravel, thoroughly consolidated and well supported beneath, may be loaded

with from 3 to 4 tons at a depth of 12 feet. The presence of a fair proportion of gravel or sand in beds of clay increases the bearing power very considerably. Except in cases where piles of wood are used in a foundation, the presence of water is distinctly prejudicial, and therefore the ground should be kept as dry as possible, by means of drainage.

Wind-pressure as affecting foundations.—The pressure upon a foundation is affected by wind-pressure in a manner similar to that upon the several portions of the shaft. But it is very undesirable that the edge of the foundation towards the wind should be completely relieved from weight. With this object, the centre of pressure should never be displaced from the centre of figure by a greater distance than the ninth part of the whole width. When the foundation is a square one, which is usually the case, this causes the load under wind-pressure to vary from $\frac{1}{3}$ of the mean pressure at the weather side to $1\frac{2}{3}$ times the pressure at the leeward side. This is almost invariably safe, as upon earth a greater load may be applied momentarily (as by reason of wind-pressure) than it would bear constantly with safety.

Uniformity of foundations.—The most scrupulous care should be taken to ensure that the character of a foundation shall be uniform throughout, or rather symmetrically disposed with regard to the weight imposed upon it. A site upon which the ground is variable in its nature is most treacherous, and should be avoided, even though on a preliminary inspection it may appear generally firm and good. Rock should be levelled, cleaned, and roughed before building upon it. Sloping ground is to be avoided where possible. Faults, landslips, and signs of movement of any kind in the ground should be most scrupulously avoided. If

even a crooked tree should be found on the ground, the cause for it should be ascertained. Defective foundations have caused trouble and loss in every respect on so many occasions, that the necessity for the greatest care with a view to safety may be considered to be established.

Exceptional foundations.—The careful application of the foregoing means will in almost every case ensure a safe foundation. But an exceptional case may arise which can only be met by a study on the spot. One interesting case was reported in *The Builder* of March 13, 1886, p. 422, in which an inverted masonry pyramid was sunk in soft ground and successfully carried a chimney when other means had failed.

Examples.—The leading principles set forth above are embodied in Figs. 17, 18, 19, 20, 21, and 22, which show a chimney of the proportions treated in the calculated example appended. The proportions are purposely left uncorrected after the first trial, the results of which show that the chimney requires to be slightly strengthened in order to withstand a wind-pressure of 56 pounds per square foot. The face from *a* to *g* is built to one uniform batter. The top length is made 1½ bricks in thickness, and the bottom length of the battered shaft is made 3 bricks in thickness, the vertical pedestal being 3½ bricks. The change in thickness at point *g* is made by means of an outer set-off in the brickwork, each of the upper changes being made in the inside.

Example with hollow batter.—An alternative chimney of the same height as the above, and with the same cap and pedestal, but built with a hollow batter, would present a better appearance. Most of the set-offs would, however, require to be made at higher points, in order to give sufficient strength as against wind-pressure. This type is chiefly to be recom-

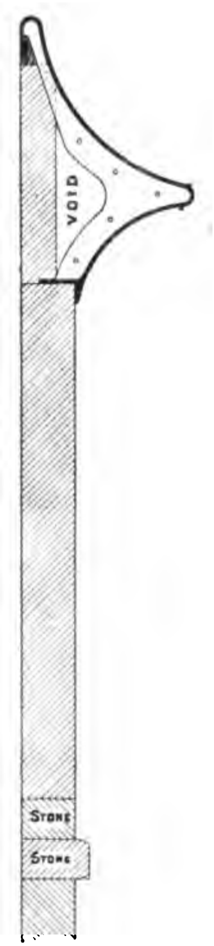

Fig. 17.—Section of chimney-cap or capital.

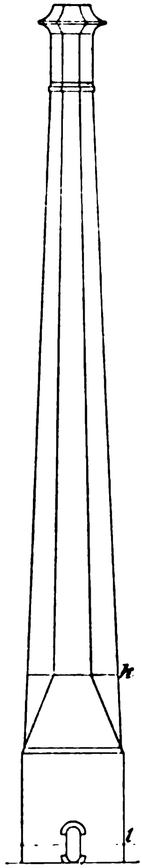

Fig. 18.—Elevation of chimney.

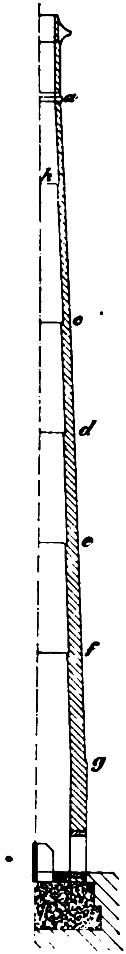

Fig. 19.—Half-section of chimney.

mended for adoption in circular shafts. Table III. gives a series of outside widths applicable to such a case, which may be varied indefinitely. Each division is assumed to possess a straight batter, but the whole would present the appearance of an unbroken curve;

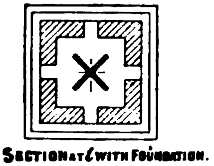

Fig. 20.—Horizontal section of base of chimney.

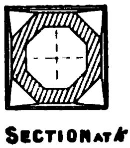

Fig. 21.—Horizontal section of chimney above pedestal.

Fig. 22.—Plan of cap of chimney.

the greatest amount of departure from a true curve is one inch at the bottom and one half-inch at the top of the shaft. No special precautions are required, beyond changing the batter rule in use at each of the points specified.

TABLE III.—SHAFT OF CHIMNEY WITH CURVED BATTER, SECTIONS OF EQUAL LENGTH.

Section.	Width.	Difference.	Increment of difference.	Batter.
	ft. in.	inches.	inches.	
a.	9 0			
		10		1 in 55·0
b.	9 10		4	
		14		1 in 40·2
c.	11 0		5	
		19		1 in 29·2
d.	12 7		6	
		25		1 in 22·2
e.	14 8		7	
		32		1 in 17·4
f.	17 4		8	
		40		1 in 13·8
g.	20 8			
		(9)		
Below *g.*	21 5			

Observations to be recorded.—Throughout all chimney-building operations, from the time of the first appearance of the work above ground to its full completion, and afterwards until its permanent stability is fully assured, an accurate and uninterrupted series of levels should be taken and recorded. These are most conveniently obtained by the use of special instruments, but the work can be performed with only a good plain spirit-level and straight-edge. For this purpose small brass blocks may be inserted in the walls, at distances to suit the length of straight-edge. These should be fitted with adjusting screws, to be accurately set to one level, and secured beyond the possibility of interference and damage. If possible, the absolute level as compared with that of some point of assured stability should be taken, but the comparative levels of the several points are of chief importance, as any settlement which takes place should affect the whole equally. In case any irregularity should develop and be found to continue or increase, the matter should not be neglected one day, as every delay increases the difficulty of treatment, even if it does not render the work dangerous. Constant checks should also be applied to the vertical centre-line of the shaft, to the batters, and to the angles or circularity, according to the form adopted.

Condition of mortar.—The work should not be carried forward too rapidly for the due setting of the mortar. In addition to a restriction upon the daily progress, an absolute stoppage of a few weeks is usually made for the benefit of the mortar on one or two occasions during the erection. This stoppage is also useful in promoting a consolidation and drying of a soft or wet earth foundation, by reason of superincumbent pressure. When piles are employed, this compression is not necessary or possible. During wet weather work should be sus-

pended, and only resumed after a careful examination of the whole has been made. If a period of continued wet weather from one quarter is followed by gales from the opposite quarter, the heavy pressure upon soft mortar causes it to yield and allow the top of the shaft to lean over. This brings over the centre of gravity, so that the weight is further increased upon the weak side; the foundations yield, and the deflection is intensified. Even when its quality is above suspicion, and it is fully set, the mortar is more susceptible to pressure than the bricks, and hence the importance of thin joints. But it is equally important that the joints should be completely filled with mortar, as any spaces which may be left cause the imposition of a correspondingly greater load upon the area covered by mortar, and therefore a greater amount of yielding or compression. It may very well occur that chimneys which are noted for special flexibility in a gale suffer from indifferent bedding of the bricks.

Upward increase in chimney opening.—A short chimney, such as those used upon steam cranes and locomotives, is improved in draft power by an increase in diameter at the top by reason of the gradual reduction in velocity of the gases. But in a tall brick chimney the velocity of draft is much lower, on account of the absence of forcing by the use of a steam jet. A few brick chimneys have been built with the inside width greater at the top than at the bottom, but there appears to be no evidence of superiority arising from such arrangement. However this may be, it is clear that the usual tapering form of chimney possesses structural advantages of sufficient force to surpass the disadvantages. By its means the weight of the top portion is so much reduced as to bring the centre of gravity of the whole to a position much nearer to the ground than

would be possible with an opposite type, thereby increasing the stability. The appearance of the top in a gale suggests the probability that the diminished velocity of the gases, in case of an increased area, would place them at a disadvantage in making their exit against the wind.

Isolation of chimney.—Wherever possible, the foundation of the chimney should be kept quite apart from all other foundations. In any case, the superstructure must stand quite alone, and not attached to any building, or utilized for the support of any object whatever.

Chimney lining.—The main brickwork of a chimney is usually protected from the heat or the chemical action of the gases by a fire-brick lining, which may extend about 10 feet upwards from the bottom, or the whole height, or to any intermediate height. The lining should not be connected to the main structure, but allowed perfect freedom of expansion. Explosions have occurred by reason of accumulation of gases between the two parts, but this risk has not been found to be an important one. Very slight means of ventilation serve to avoid the risk if there is surplus draft power sufficient to bear a little air leakage. The intervening space is sometimes filled with sand, but this should not be allowed to extend more than 20 or 30 feet from the bottom, owing to the bursting pressure which may be imposed upon the brickwork. The lining should be built of fire-brick set in fire-clay, and may usually be of uniform thickness throughout. In connection with the service of steam boilers, there is seldom any necessity for carrying the lining to a great height. But in chimneys connected with chemical works, a lining of the entire height would often be most useful in protecting the mortar in the outer brickwork from the attack of acid and other chemical fumes, in which case the

annular space should be more freely ventilated. The chimneys of such works give more than an average amount of trouble. This arises largely from the total or partial destruction of the mortar. But in some cases there appears to be good reason to suspect soakage into the foundations of substances whose composition and action upon the soil are beyond all powers of estimation.

Disintegration of substance.—Precautions should be taken against the destructive action of the weather upon a chimney, as upon any other structure. If this is not done, rain-water soaks in, becomes frozen, and causes rending of the structure on a large or small scale, or disintegration and exfoliation of substance. Heavy rain, driven against a chimney in a strong gale, washes away any material which may have been disintegrated. This process, often repeated, causes the face of the mortar in the joints to recede, and each brick to show a well-developed ledge, upon which water may collect. If, under such conditions, the bricks are at all liable to absorb water, this is sure to occur, the waste will increase, and the strength and general condition of the chimney will suffer accordingly. Such action is very much increased by porosity in the mortar due to over-sanding. Rain-water also possesses a solvent action upon the lime in mortar, so that it is washed out, leaving sand, which rapidly follows the lime. Mortar which is made from good Portland cement and sand, in suitable proportions, suffers exceedingly little from the weather. That from lias lime suffers more, but still not very seriously, as a rule. Mortar made from lime derived from carboniferous limestone usually suffers very much when freely exposed, and chalk lime is unfit for use in the work under consideration. Porous bricks and bad mortar help each other to absorb moisture, so that the

ruin of the two proceeds rapidly, when either would survive under more favourable surroundings.

Protection by cap and by pointing of joints.—For the protection of the structure against the weather, the first condition is a weather-proof cap. No arrangement of cap in which bricks of ordinary size are used should be allowed. All joints must be well cemented and grouted solid throughout. If the shaft is built with good lias or hydraulic lime mortar and carefully finished as it is built, further attention may possibly be dispensed with. But generally this is not the case, and the shaft will require to be cleaned down and pointed after the completion of the cap. This is best effected by means of cement mortar or by hydraulic mortar, which is improved by the addition of iron forge scale. In either case the volume of sand employed should be from two to two and a half times as great as that of the cement or lime, with a view to prevent shrinkage or fine cracking on the one hand, and on the other to secure strength and impenetrability to moisture. The pointing should be carried over the whole of the cap inside and outside, the outside of the shaft and of the pedestal, beginning at the top and proceeding downwards. Before the insertion of pointing material, each joint should be perfectly cleaned out by scraping and brushing to a depth of from three-quarters of an inch to one inch; and the material inserted by tight ramming. The finished surface of the pointing should be left smooth and free from ledges upon which moisture could collect, either upon the pointing or upon the brick. In pointing brick-work the face of which is vertical or nearly so, the pointing may, if desired, be formed with an over-hanging lip at the lower edge, slightly throated, so as to shed the water away from the face of the brick. The importance of trustworthy attention to

every detail involved in this operation cannot be over-estimated.

Protection of ledges by slate.—A covering of slate is often applied with advantage to projecting parts of brickwork, as in the pedestal of a chimney. This prevents soakage of rain-water into the structure. The upper edge should always be made tight by flashing, or much of the possible benefit is lost.

Abnormal cleanliness of surface.—When the joints of brickwork, or the bricks themselves, collectively or individually, appear to be exceptionally clean, waste or exfoliation of substance is generally proceeding. The same result may however, in rare cases, be observed to be due to exceptional density of surface preventing the adhesion of organic or inorganic foreign matter.

Structural cracks.—A chimney in use often develops rents, as a result of weakness or of exposure to high temperatures. A rent in a horizontal direction across a chimney is very rarely, if ever, formed. Such would indicate that the weight of the chimney is insufficient to oppose the overturning power of the wind, and would call for immediate steps to verify the fact and take appropriate action. Rents running diagonally across a chimney are occasionally met with, and are chiefly, if not entirely, due to weakness, which may have originally existed in the structure, or may have arisen as a consequence of departure from a vertical condition. Any such signs should receive immediate attention, or the damage will almost certainly extend. If the chimney leans over, it may possibly be restored to a vertical condition by cutting out one side; but as to a chimney which has been subjected to this operation, some doubt must always exist as to the uniformity of pressure in the brickwork in proximity to the part

treated. A chimney in this rent condition should generally be shortened, and the cracked part either entirely removed or secured by bolts or straps. Rents in a vertical direction are more frequent than any others, especially in chimneys exposed to varying or high temperature. The strength of a chimney for opposing wind-pressure is much affected by vertical rents, just as two separate beams, placed in contact one above the other, are less strong than one of double size. For this reason such cracks should be secured from extension by straps or bolts, encircling the chimney above and below the crack. Intermediate ones should also be placed across the crack, so as to secure the two parts together, and prevent abrasion in windy weather, by which the opening would be increased. The mere holding of the two parts into close contact also materially increases the strength of the chimney.

Iron strap-ties.—Flat bar-straps of wrought-iron or steel, held together by bolts at each junction, are often used for chimney-ties. These seldom fail by breaking across the body, but the bolts break, or the bolts or the lugs are strained, so that very often the strap

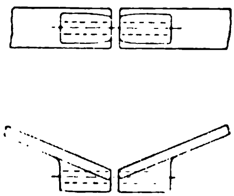

Fig. 23.—Lugs of strengthening straps for chimney.

becomes non-effective. With a view to avoid this, the ends of the straps should be made to the form shown in Fig. 23, or even with a longer bolt cut slightly into the brickwork, and the bosses modified accordingly.

The example shown is for use upon an octagonal chimney, but the same principle applies to square or circular chimneys.

Bolt-ties.—Round bolts may be used for binding square or octagonal chimneys. These require special plates at the angles, of which a form adapted for use in soft cast-steel is shown in Fig. 24. Such plates should be well bedded against the brickwork by means of cement. The hole in the centre of the plate will be

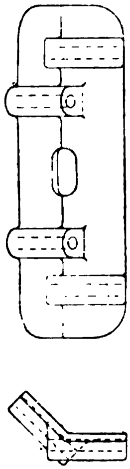

Fig. 24.—Angle-plate for strengthening bolts for chimney.

found to be of assistance in securing a good bearing near the bolts, and avoiding excessive bearing in the centre. The hole may be finally filled up straight, to form a key to prevent the plate from slipping down. Such plates may be thinned at the edges to reduce weight, but this should not be carried far. They require to be made right- and left-handed, with the bosses set to suit the batter of the chimney. The plates may be spaced

at any distance vertically apart to suit the case in hand. The bolts should be tightened very gently and repeatedly, so as to ensure uniformity of bearing throughout. This should be done while the chimney is at work, otherwise the bolts may become subject to a dangerous degree of tension when the brickwork becomes expanded by heat. All cracks should be pointed and grouted with neat cement. The best time for doing this is when the bolts are partially tightened up, after which the cement should be allowed to set and the bolts completely tightened.

A chimney which requires to be bound with iron-work must always be carefully treated, and guarded against exposure to extreme temperatures, especially in unsettled weather, or a gale may strike it at a disadvantage, either when the ironwork is slack or when it is under excessive tension. Ironwork used for such a purpose should be carefully protected by cement, tar, or paint, for the prevention of corrosion. If before the nuts are run on the screw-threads are smeared with a little tar, neutralized with quicklime, they will better resist the weather. The bolts should be provided with deep nuts, and all the better if with concave spherical washers, to ensure fair bearing. The bolts should be sufficiently long to show three threads outside each nut. These threads should be ultimately filled with tar to protect the whole, and prevent slacking back of the nuts. The whole of the ironwork used for this purpose should be subjected to thorough examination at intervals depending upon circumstances.

Access for examination.—In connection with all chimneys specially subject to deterioration, it would be well to provide some facility for easy access to the top for purposes of examination. Such a provision might take the form of gun-metal or muntz metal bars built

into the chimney, at spaces about 9 feet apart or to suit short existing ladders, which could be secured to the bars and the ascent made in a very short time. Such provision should be made in proximity to the lightning-conductor, so that an examination of the same could be made conveniently, but not so as to involve probability of damage to the conductor.

Lightning-conductor.—The lightning-conductor of a chimney should be of large sectional area, with a good connection to damp earth, and carried to the top without abrupt bends. The finial should be of metal suitable to withstand corrosion, and the terminals protected by a platinum coating. Each terminal should be brought to a sharp point, and it is better that the rod should be carried through the chimney-cap (properly protected), rather than around the moulding.

Precautions to be adopted in the use of iron.—Iron should never be allowed to be built into a chimney if it can be avoided, on account of its liability to corrosion, in which process it develops a very strong expansive force, which sometimes lifts or bursts the work and causes great damage. For this reason the most scrupulous care is necessary to prevent the access of moisture to any bond-iron which it may be considered necessary to use. Iron steps or other fittings should never be allowed to be built into the brickwork inside the chimney.

Danger arising from neglect.—A consideration of the consequences of neglect, in risk to life, damage to property, disarrangement of work, and other losses, will show that chimneys ought to receive much more attention than they very often do receive. Perhaps one reason for this is the common knowledge that structures which lean very much away from the perpendicular exist in different parts of the world, and

have so existed for ages. But it will be found that the circumstances under which these exist are not comparable with those appertaining to chimneys. It is not too much to say that in every case the amount of slope as actually *measured* is much less than is supposed, owing to the unconscious exaggeration in such matters to which artists are subject. In the majority of such cases, the proportions of width to height are very much greater than in chimneys. In each case the condition and arrangement of foundation and the history of the structure is involved in mystery. None of them are exposed to the temperature and fumes imposed upon the foundation and superstructure of a chimney. Perhaps, also, differences of climate may form a factor of importance. On the whole, therefore, such instances have no bearing upon present practice, and ought to be dismissed from consideration.

TABLE IV.—CHIMNEY CALCULATIONS.

SHEET 1.—CHIMNEY 180 FEET HIGH. GENERAL PARTICULARS. FIRST TRIAL.

Concrete foundation, 28 ft. square, 12 ft. below ground.
Brickwork pedestal, 21 ft. 5 in. square, 23·10 ft. above ground.
Octagonal shaft, tapering from 20 ft. 8 in. in outside width to 9 ft. Shaft, 138·90 ft. in length.
Capital, 9 ft. wide outside brickwork, 6 ft. 9 in. inside, 18 ft. high.

Area of opening at top = $81 \times 81 \times \cdot 8284 = 5435$ sq. in.
5 boilers, each burning 720 pounds of coal per hour, require, under favourable conditions, an opening = $5 \times 720 \times 1\frac{1}{2} = 5400$ sq. in.

Section *d*: area 93·8 sq. ft., load 478,677 lbs. = 2·28 tons per sq. ft. mean pressure, increased to 4·56 tons by wind 53·5 lbs. per sq. ft.
Section *f*: area 139·8 sq. ft., load 1,081,707 lbs. = 3·45 tons per sq. ft. mean pressure, increased to 6·90 tons by wind 57·2 lbs. per sq. ft.

Foundation, 28 ft. sq., 12 ft. below surface, load 1452 tons, mean pressure 1·85 tons per sq. ft.
Maximum displacement of centre of pressure from centre of figure = $\frac{28}{9} = 3{\cdot}11$ ft. Load varies from 3·08 to ·62 tons per sq. ft. during wind 53·6 pounds per sq. ft.

TABLE V.—CHIMNEY CALCULATIONS.

SHEET 2.—CHIMNEY 180 FEET HIGH. DETAILS OF WEIGHT. FIRST TRIAL.

Divisions.	Outside widths.	Mean outside widths.	Inside widths.	Lengths.	Contents.	Weight of each division.	Load on each section.
	feet.	feet.	feet.	feet.	cubic feet.	pounds at 112 lbs. per cu. ft.	pounds.
Capital	9·00	9·00	6·75	18·00	528·5	59,192	59,192
a. h.	9·00	9·72	7·47	17·15	549·5	61,544	120,736
h. c.	10·45	11·66	8·66	29·15	1,472·1	164,875	285,611
c. d.	12·88	13·86	10·11	23·15	1,723·8	193,066	478,677
d. e.	14·83	15·80	11·30	23·15	2,338·6	261,923	740,600
e. f.	16·78	17·75	12·50	23·15	3,045·6	341,107	1,081,707
f. g.	18·73	19·70	13·70	23·15	4,341·6	486,259	1,567,966
Pedestal (sq.)	20·67	21·42	14·67	23·10	5,627·4	630,269	2,198,235
Foundation		28·00		12·00	9,408·0	1,053,696	3,251,931

TABLE VI.—CHIMNEY CALCULATIONS.

SHEET 3.—CHIMNEY 180 FEET HIGH. DETAILS OF WIND-PRESSURE. FIRST TRIAL.

Effective arm of wind-pressure, each division.
Moment of wind-pressure, each division.

Section.	Effective pressure upon each division, by reason of wind striking perpendicularly with force of one pound per square foot. 21·42×23·1 =494·80	19·70×23·15×·7 =319·24	17·75×23·15×·7 =287·64	15·80×23·15×·7 =256·04	13·88×23·15×·7 =224·60	11·66×29·15×·7 =237·92	9·72×17·15×·7 =116·69	9·00×18·00×·7 =113·40	Total moment of wind-pressures at each section, foot-pounds.
a.								9·00 1,020	1,020
h.							8·58 1,001	26·15 2,965	3,966
c.						14·58 3,469	37·73 4,403	55·30 6,271	14,143
d.					11·58 2,601	37·73 8,977	60·88 7,104	78·45 8,896	27,578
e.				11·58 2,965	34·73 7,800	60·88 14,485	84·03 9,806	101·60 11,521	46,577
f.			11·58 3,331	34·73 8,892	57·88 13,000	84·03 19,992	107·18 12,507	124·75 14,147	71,869
g.		11·58 3,696	34·73 9,990	57·88 14,820	81·03 18,199	107·18 25,500	130·33 15,208	147·90 16,772	104,185
Ground line	11·50 5,690	34·68 11,071	57·83 16,634	80·98 20,734	104·13 23,388	130·28 30,996	153·43 17,894	171·00 19,391	145,798
Foundation	32·50 16,081	55·68 17,775	78·83 22.675	101·98 26,111	125·13 28,104	151·28 35,992	174·43 20,354	192·00 21,772	188,864

TABLE VII.—CHIMNEY CALCULATIONS.

SHEET 4.—CHIMNEY 180 FEET HIGH. DETAILS AFFECTING STRESSES UPON SECTIONS. FIRST TRIAL.

Section.	Outside width.	Thickness of wall.	$\frac{t}{w}$	Admissible displacement of centre of pressure.	Moment of wind-pressure from Sheet 3.	Load from Sheet 2.	Corresponding wind-pressure.
	feet (w).	feet (t).		feet (d).	foot-pounds (m).	pounds (l).	pounds per sq. ft. $\frac{l \times d}{m}$
a.	9·00	1·12	·124	1·87	1,020	59,192	108·5
h.	10·45	1·12	·107	2·24	3,966	120,736	68·2
c.	12·88	1·50	·116	2·72	14,143	285,611	54·9
d.	14·83	1·88	·126	3·08	27,578	478,677	53·5
e.	16·78	2·25	·134	3·44	46,577	740,600	54·7
f.	18·73	2·62	·140	3·80	71,869	1,081,707	57·2
g.	20·67	3·00	·145	4·18	104,185	1,567,966	62·9
Ground line	21·43	3·38	·158	4·28	145,798	2,198,235	64·5
Foundation	28·00	Solid	Solid	3·11	188,864	3,251,931	53·6

APPENDIX TO CHAPTER XIV.

Calculations upon the design of chimney 180 *feet high.*

SHEETS 1, 2, 3, AND 4.

ALL leading particulars are based upon the results of preliminary work, to be subjected to the first set of calculations. Details to be modified in accordance with the results of first calculation, and the second set of calculations completed as a thorough check.

Sheet 1 gives the chief particulars, and demonstrates the sufficiency as to power.

In Sheet 2 the details of the weights are given. The cubic contents of the several divisions of the tapering shaft are based upon the sectional area at the centre, multiplied by the length of division. This gives the cubic contents very slightly below the correct value, which can only be obtained by more tedious calculation. By assuming the mean sectional area of each division to be equal to the arithmetical mean between the areas at each end, the error is a slightly greater one in the opposite direction. The bricks are assumed to be of such dimensions that the thickness of each wall will be a multiple of 4½ inches. The weight of brickwork is assumed to be 112 pounds per cubic foot.

In Sheet 3 the details of wind-pressure are given, on the assumption that the pressure of wind equals one pound per square foot of surface upon which it strikes perpendicularly. At the head of the table the effective

pressures upon the several divisions are given. In the pedestal this is equal to the area directly opposed to the wind. In the upper divisions the area is reduced by a factor which is in this case ·7, the shaft being octagonal. The proper factor for circular shafts is ·5. In the body of the table two figures are given in each place. The upper figure gives the distance in feet from the section to the centre of one of the divisions above it. The lower figure gives the product of the first multiplied by the figure at the head of the column, and is the moment expressed in foot-pounds, of the pressure of wind upon the division, and as affecting the section in question. The total moment upon each section as given in the last column is the sum of all those in line, or of the action of the wind upon all divisions above the section in question.

In Sheet 4 the width of chimney and thickness of wall at each section are given; also the ratio which the latter bears to the former. The admissible displacement of the centre of pressure from the vertical line through the centre of gravity of the whole mass above the section in question is obtained by the use of a fraction corresponding to the ratio $\frac{t}{w}$ and the form of the chimney. This is obtained from Fig. 16, and being multiplied by the width of section gives the admissible displacement. The moment of wind-pressure is transferred from Sheet 3, and the load upon each section from Sheet 2. The safe wind-pressure at each section is then obtained by multiplying the load by the admissible displacement, and dividing by the moment of wind-pressure.

In calculations made upon a leaning chimney, the safe wind-pressure upon the whole or any part is similarly obtained, after deducting from the admissible displacement the amount by which the centre of gravity of the part in question departs from the perpendicular. In every case full notes should be recorded on the face or back of the sheet.

In the first trial it will be noticed that the part which

would fail by overturning with the lowest wind-pressure is at section *d*. But at this point it is quite safe under a wind-pressure of 53·5 pounds per square foot. Here the mean pressure upon the brickwork is 2·28 tons per square foot, which may be noted upon Sheet 1. During the prevalence of a wind of this pressure the brickwork upon the leeward side of the chimney will be loaded to 4·56 tons per square foot, and that upon the weather side will be just relieved from pressure. The corresponding pressures upon section *f* are 3·45 and 6·90 tons per square foot under a wind-pressure of 57·2 pounds per square foot. The latter pressure is greater than should be imposed upon inferior brickwork, but is quite safe upon good brickwork.

In the example, the wind-pressure has been assumed to be freely applied to the chimney from base to cap. This can only arise on a bare hill-top. Chimneys are often erected in such situations to take advantage of rising ground and save height in the shaft. When accurately proportioned, they are usually of stout form and therefore sufficiently safe. When, however, a chimney is erected near to high buildings, the wind-pressure is more or less deflected from the lower divisions. The protection afforded in this way is obvious as regards a gale blowing over the building towards the chimney. Though less obvious, the protection afforded against a gale blowing in the opposite direction is very effective. But as to wind from other directions no rule can be given.

Important works sanctioned by the Board of Trade for erection in highly-exposed situations, such as the Forth Bridge, must be constructed for safely withstanding a wind-pressure of 56 pounds per square foot. This is probably never exceeded in the British Islands, at least not over the area exposed by a chimney. In districts closely built upon, the force of the wind is very much checked. The amount of such reduction is very difficult to estimate, but that it is one of some importance is constantly demonstrated in the examination of roofs and chimneys, by the occurrence of loose

details which could never resist a wind-pressure of 56 pounds.

The mean pressure upon the foundation is 1·85 tons per square foot, which, as already explained in the chapter, is quite safe in good ground.

In complete calculations, the possibility of sliding must be considered. The co-efficient of friction of fresh brickwork is about ·5, and that of set brickwork about 67, apart from any possible strength of joint. A straight wall of brickwork one foot in thickness, of any height, would be just on the point of sliding along before a wind of 56 pounds per square foot applied at right angles to it. A square chimney of any height or width one-quarter of a foot in thickness would move similarly. Also an octagonal chimney two-ninths of a foot in thickness, and a circular one of one-sixth of a foot in thickness. In each case the corresponding thickness of fully-set brickwork is one-fourth less. Obviously this element is not a serious one, as affecting the main structure of a chimney. But it is of importance as affecting the use of very light caps, or loose details in connection therewith.

If desired, Sheets 1, 2, 3, and 4 may be carried through simultaneously, especially in the last set. Each and every sheet should be numbered as to its set, and dated. The last set should be distinctly marked *final*.

CHAPTER XV.

ECONOMIZERS OR FEED-HEATERS.

Primary duty of economizer.—In most cases in which large powers are dealt with, the feed-water, on its way from the pump to the boiler, is passed through an "economizer," the primary object of which is to raise its temperature. This object is attained by the withdrawal of heat from the furnace gases, after they have left the boiler. In the usual form of the apparatus, as introduced by Sir Edward Green in 1849, a battery of vertical cast-iron pipes is employed; these are connected by top and bottom boxes, which again are connected by top and bottom branch-pipes, in such a manner that the total length of pipe to be traversed by the water, from the supply-branch to the delivery-branch, is practically equal, whichever pipe the water may follow in its course. By this means, the distribution of the water throughout the apparatus is effected with a satisfactory degree of uniformity. The furnace gases are led amongst the pipes on the way from the boiler to the chimney in ordinary work, an alternative flue being provided for use when access to the economizer is required. In the arrangement of the flues, the same care should be taken to give easy curves and an air-tight condition, as in connection with the boiler setting. The area of cross-

section of the flue should be sufficiently large to avoid excessive resistance to the air-currents, when an accumulation of soot occurs, but it should not be greater than necessary; it should be as uniform as possible, and dead-ends and irregular spaces avoided. Access to the lower ends should be provided, conveniently for reaching all covers and valves, for cleaning operations and inspection. Space should be provided on one or both sides of the series of pipes, for a man to pass along; such space to be closed by deflecting dampers, to throw the hot gases among the pipes and prevent failure in the function of the economizer. It is an advantage to have the conditions of these dampers, as to being open or closed, clearly indicated outside the brickwork. The pipes and boxes require washing out, at periods dependent upon the condition of the water used; the waste to be led away by a pipe, so as to avoid soaking into the foundations. This is sometimes difficult to effect, by reason of the levels of existing drains, but the point is important, and a large quantity of water is required in the work. In an extreme case, the provision of a pump is necessary.

Removal of matter collected on or in pipes.—The pipes become covered on the outside by soot, the amount of which is in proportion to the amount of smoke produced in the fires. If allowed to accumulate to an appreciable thickness, such a coating of soot would interfere seriously with the transmission of heat to the water. The coating is therefore removed by reciprocating scrapers, continuously operated by the main engines or by a small separate engine; sometimes the feed donkey-engine is utilized for the purpose. When the feed-water contains much solid matter, and especially if blowing-off of the economizer is neglected, the pipes become coated inside. This coating may become so extensive as to require the application of a boring process for its removal. Valves

are provided on the inlet and outlet branches for shutting off the water, when the gases are diverted from the pipes. A blow-off valve for scouring and emptying the pipes is provided, and a safety-valve for the relief of any pressure which may arise in the pipes.

Improvements in apparatus.—The general form of the apparatus has remained practically unchanged since its introduction, but important improvements have from time to time been made in the form and manner of driving the scrapers, in the covers provided for clearing the pipes, in the means of access to the several parts of the apparatus, in the means adopted for giving facility for expansion, and in improved means of manufacture, especially with a view to ensure strength and uniform soundness in the pipes and all other parts. Facility for expansion is more urgently required at the present time, on account of the high temperatures to which the apparatus is exposed, and its greater dimensions. The pipes are also now arranged so that on the rare occasions when one bursts, it may be replaced very quickly, without involving any disturbance to neighbouring parts, this being facilitated by the practice of jointing the pipes, metal to metal, closed by forcing.

Purification of water in economizer.—Though the apparatus is generally adopted from direct motives of economy, yet it very frequently performs most useful work in the removal of solid matter from the feed-water, especially in cases where hard untreated water is employed, the solid matters becoming separated in the pipes, by reason of the application of heat. Owing to the fact that the economizer is exposed to flue gases of a temperature lower than those to which any part of the boiler is exposed, the presence of solid matter is less objectionable; it is also more conveniently removable. Hence the use of the economizer as an

agent for the purification of water is by no means unimportant.

Limits of economy.—As an agent for the promotion of economical working, the power of an economizer varies exceedingly under different conditions. It is not found practicable to use an economizer for the actual generation of steam, which operation must be confined to the boiler. Any amount of heat may, however, be imparted to the water in the economizer, up to the point of raising steam. It therefore follows that of the whole amount of heat expended in the production of steam, only the sensible heat can be imparted in the economizer, while the duty of imparting latent heat must be confined to the boiler. The exact proportion of the two will vary with the temperature of the feed-water before entering the economizer, and with the temperature of the water after leaving the economizer The latter is limited by the pressure in the boiler; with higher pressure a higher temperature in the economizer becomes possible, and thus greater advantage becomes possible than with low pressure. Table VIII. gives the maximum possible percentage of economy, which can be directly secured by the use of an economizer. In very few cases does the temperature of water fed to boiler approach that of the steam; therefore the percentage of economy actually obtained is usually below the corresponding one in the table, though in some cases it closely approaches such figure. Higher percentages of economy are sometimes secured by the use of an economizer, by the simple relief of an overtaxed boiler, which is fired so hard that the heating surfaces of the boiler cannot absorb such a large proportion of the heat generated in the furnace, as they would do if forced to a rather less degree, as for instance would be the case if the boiler power were itself increased.

TABLE VIII.—AVAILABLE PERCENTAGES OF ECONOMY AT DISPOSAL OF ECONOMIZER.

Temperatures at which water enters the economizer.	Pressures of steam above atmosphere—pounds per sq. in.			
	50	100	150	200
60° F.	21·0	24·3	26·6	28·4
90° F.	18·8	22·3	24·7	26·5
120° F.	16·6	20·2	22·6	24·5
150° F.	14·1	17·9	20·5	22·4

Proportions.—A good proportion of pipes for use in connection with Lancashire boilers is attained when the heating surface of the economizer is about equal to, or a little greater than, the total heating surface in the boilers in use at one time. Boilers with less heating surface than Lancashire boilers, or which are unable to extract so much heat from the gases and are working at a high pressure, may with advantage be provided with a larger number of pipes, but the production of steam must be avoided. Each square foot of economizer surface should absorb 900 to 1200 units of heat per hour. Usually the temperature of the feed-water is increased by 190° to 220°, when each square foot of economizer surface suffices to treat 4 to 6 pounds of water per hour.

Physical relief to boiler by use of economizer.—In all cases the supply of hot water to the boiler is advantageous, both as affecting the total economical result, and as giving relief to the boiler from stresses due to differences of temperature. In the same way, the economizer itself is relieved very much if the water is

supplied to it at a high temperature. An important service often performed by an economizer is the heating of water supplied to a boiler preparatory to raising steam after cleaning, or for other purposes; this should never be omitted if possible, as by this means the boiler is heated with a very much greater degree of uniformity than by the application of a fire after filling with cold water.

Precautions to be observed.—Best results from the use of an economizer can only be obtained by careful attention to blowing-off at the scouring-valve and at the safety-valve, by frequent cleaning of pipes and soot chamber, and by avoiding in-drafts of cold air. The last occur by reason of defective brickwork, the remedy for which is obvious; also at the holes for the admission of scraper-chains, which should be closed by blocks accurately shaped to fit the contour of the chains. In some cases the holes are automatically sealed fairly completely by tar, grease, and dust. Dampers should be provided and operated as described in connection with boilers.

Application to feed-heaters of exhaust steam from engines.—In connection with non-condensing engines, the exhaust steam is often used for heating the feed-water. In some cases the two are mixed together, in a vessel very similar in its essential features to an ordinary injection condenser without air-pump. In connection with this type, it is necessary that the feed-pump should be adapted for dealing with water at a high temperature. In other cases, the water, on its way from the feed-pump to the boiler, is passed through a worm, or a nest of tubes, or a series of inverted U-tubes, placed in what is practically an enlargement of the exhaust-pipe. In the first type of heater, the water loses some of its impurities, suspended and dissolved, but on the other

hand is very apt to absorb an objectionable amount of greasy matter, carried from the cylinders by the current of steam. In the second type, the feed-water is protected from greasy contamination, but loses all advantages as to the separation of any impurity already contained. Obviously, the suitability of one type over another must depend upon the special conditions of each particular case. The maximum percentage of economy to be secured in any case may be obtained =

$$\frac{100 \times \text{sensible heat acquired in rise from feed-temperature to } 212^\circ}{\text{Total heat acquired in rise from feed-temperature to production of steam.}}$$

The quantities of heat for this purpose may be obtained from Table X.

CHAPTER XVI.

COOLING-PONDS AND APPLIANCES IN SUBSTITUTION.

Quantity of heat imparted to condensing water.—It is elsewhere explained that from 80 to 90 per cent. of the heat imparted to the steam in the boiler re-appears in the condensed water discharged by the air-pump. The heat thus incorporated with the discharged water causes its temperature to rise so much that it is quite unfitted for further use in effecting condensation, until it has lost such additional heat. If a continuous supply of cold water is not available, a cooling-pond or an equivalent device is thus a necessary adjunct of a condensing engine.

Dispersion of heat acquired by condensing water.—In an effective cooling-pond the heated water is exposed to the air, over a surface sufficiently large to allow the rejected heat to be dissipated as rapidly as it comes from the engine. This heat is disposed of in four directions—

I.—By the action of the atmosphere in causing evaporation from the surface, whereby heat is rendered latent, practically the whole of which heat is abstracted from the water in the pond.

II.—By pure radiation into space.

III.—By direct conduction of heat from the water to the air.

IV.—A small and uncertain amount disappears by conduction into the earth.

Dispersion of heat by evaporation involves loss of water.—If the whole of the rejected heat were dispersed through the instrumentality of evaporation, the quantity of water lost from the surface of the pond would bear a very large proportion to that evaporated in the boiler. It would be subject to reduction on account of the 10 or 20 per cent. of the total amount of heat which is converted into mechanical work in the engine. One pound of water evaporated at ordinary temperatures from a pond will carry off less heat by 7 or 8 per cent. than one pound of water evaporated in the boiler. Combining the two corrections, it will be found that the quantity evaporated from the pond will be rather less than that evaporated in the boiler. But in many cases the loss of water which takes place by reason of evaporation is a matter of importance, and consequently it is necessary to reduce the amount of evaporation and promote the dispersal of heat by other means. The first condition is to provide such an area, that under ordinary conditions the cooling will not fall into arrear. If the heat is not carried off as rapidly as it is supplied, the temperature will rise excessively. An increase in temperature causes an increase in evaporation, and consequently in loss of water by evaporation.

Dispersion of heat by radiation.—In the dispersal of heat by pure radiation, no water is lost. In the formation of dew, and in other ways, it is shown that a clear atmosphere is necessary to freedom of radiation. Therefore any cause which tends towards the formation of a cloud of vapour over a pond is to be

avoided. Cooling by radiation also possesses an advantage in respect to purity of water in the pond, owing to diminished necessity for making up with fresh water, which always brings in solid matter. If a continuous stream of fresh water in excess enters the pond, the overflow is diminished by the amount evaporated, so that a condition of stability may be ultimately reached in which the amount of impurity in the water in the pond bears the same proportion to that in the water supplied, as the quantity of water supplied to the pond bears to that passed away by the overflow.

Advantages of a free exposure of pond.—A free exposure on all sides to the action of the wind upon the water-surface is most advantageous. Primarily, it tends to promote evaporation, but it increases direct cooling by air contact to a still greater extent. Suspended vapour is much less extensively formed, and is promptly removed when it is produced; therefore the amount of radiation is increased. It is therefore important that all buildings, trees, shrubs, &c., should be kept at a considerable distance from a cooling-pond, especially when its area is limited, in comparison with the duty required from it. Boundary-walls should be kept as low as possible, and if required may be surmounted by open fencing. The earthwork banks of the pond should be kept as low as can be considered safe under the conditions of the case. If the crest of the bank is gently rounded, the wind will tend to follow its surface and to impinge upon the water-surface; while otherwise—and especially if the face of the bank is steep and high—the wind will shoot over the pond at a distance above the water-surface. As regards its effect upon the direction of the wind, the cross-section of the bank of a cooling-pond should be designed with an object precisely opposite to that in view in the design of a chimney-top.

Dispersion of heat by direct transfer to air.—The dispersal of heat by direct conduction from the water to the air proceeds at all times. As in radiation, no water is lost. At low temperatures the amount of heat dispersed by radiation and by direct conduction to air predominates. In ordinary cases, this condition prevails for the greater part of the time, but is reversed during the prevalence of high atmospheric temperatures.

Dispersion of heat by conduction into ground.—The dispersal of heat by conduction into the ground is probably very seldom of sufficient amount to give the question much importance. Except when the freezing temperature is approached, the coldest water is always at the bottom of the pond, and convection cannot be of any assistance in such a transfer. The exact value of this item is almost impossible of estimation.

Influence of weather upon cooling.—The weather has a great influence upon the amount of cooling which can take place in a pond. It is very tardy in calm weather, when the air is damp, and brisk when a fresh dry wind is blowing. On the occasional occurrence of calm damp weather, some dependence must be placed upon the cooling which takes place by night. The radiation which occurs on clear nights is a powerful factor in cooling and in attracting moisture from the atmosphere as evidenced by the operation of the water-pits on dry hills, which are referred to by Gilbert White, and which are still in common use. But when any considerable amount of reliance is placed upon night cooling, it follows that the temperature of the water will rise towards evening, and an objectionable amount of irregularity in working will be experienced.

Quiet entry of return-water into pond.—The heated return-water from the engines is almost invariably delivered into the pond at some distance above the

surface of the water, so that the condition of the stream may be observed at any time. In this way, also, the air-pump and pipes need not stand full of water when the engine is at rest, and so they escape danger of accident during frosty weather. The water should, however, not be allowed to fall from a height directly into the pond, as this is liable to raise the mud from the bottom, and mix it with the water. It is therefore much better to discharge the return-water upon a rough stone apron, from which it may flow steadily into the pond. By this means, also, the hot water is kept in its natural position at the surface of the pond, so that the cold water may be drawn from below.

Withdrawal of water.—The water for the supply of the engine should be withdrawn from the pond at some distance below the surface, where it is comparatively cool. But the collection of mud from the bottom should be avoided. For this purpose a vertical trunk of wood or iron may be erected, reaching from the bottom of the pond to a little way above the surface, and arranged with an open side, closed by a shutter provided with an opening, which may be adjusted as to the depth from which water may be withdrawn. A horizontal pipe or open trough may be brought from the vertical trunk as required, and carried through the bank at as high a level as practicable, having regard to the level of the water in time of drought. A syphon may be carried through the bank in the same way. In either case the possibility of leakage is reduced by placing the conduit at a high level, and at the same time any necessary repairs are facilitated. A grating should be fitted over the inlet opening to prevent the entry of rubbish, and a stop-valve or sluice provided in some convenient position.

Supply of fresh water.—The manner in which fresh

water is supplied to the pond should be such as to avoid disturbance of mud. Except for special reasons, the pipe or conduit should be passed over the bank, and not through it.

Overflow.—A bye-wash or overflow is generally necessary, to prevent the water from rising to such a height as to cause danger or inconvenience. This must be floored with hard material so as to prevent cutting down the bank when the strongest stream is flowing over it, and leakage through it should be prevented.

Scouring or drain-pipe.—A pipe for use in emptying or scouring the pond should be provided, so arranged as to drain the lowest point, and provided with a valve at the inner end. The outer end of the pipe should discharge over a paved floor, so as to avoid scouring away the foot of the bank. Such pipes are often made of insufficient area, so that an excessive length of time is occupied in emptying the pond.

Construction of banks.—Where possible, the site of a cooling-pond will be arranged upon a clay subsoil, with a view to water-tightness. If there is the least suspicion as to the perfect continuity and tightness of the subsoil, it should be thoroughly bored, examined, and made good. If the earth inside the pond is fairly solid, it may be used in the construction of the banks. A puddle core must be keyed into the hard solid clay bed, and carried up the centre of each bank. This must be made of the best clay available, chopped, watered, and puddled, and covered over with sufficient earth to prevent cracking in dry weather. Great care must be taken to make this good at the end of each day's or week's work, at the junctions of banks, at the bye-wash, and where the engine-supply and main drain-pipe pass through the bank. In very high banks brickwork is necessary for the last two purposes; but usually pipes

carried through the bank with solid flanges in the puddle to prevent creeping of water along the pipes are sufficient. All pipes carried through a heavy bank should be of corresponding strength, and should not be jointed in the puddle, but carried solid to each side of it, where a strained joint would be less likely to arise, of less consequence in case it should arise, and more accessible for repair. In many cases the drain-pipe may be laid with advantage in absolutely solid ground.

Water-tight floor.—If the subsoil is not quite water-tight, the puddle-trench must be carried down to a bed of sufficient stability, and the whole of the floor puddled and connected to the puddle-core of the banks. In the whole of these operations the most unremitting care is necessary, so that no defect may arise, as even a small original defect may lead to serious consequences, and a slight leakage is often very difficult to localize.

Protection of banks.—The banks require some protection against the action of the water in rough weather. This may be provided by means of stone or brick pitching, or in some rare cases by boarding. This protection should be sufficient to withstand the action of any ice which may be allowed to form in winter. With good ordinary slopes of earthwork, the hard pitching need not extend more than 2 feet below the level of water-surface. The slope usually adopted is about 1 vertical to 2 horizontal, steeper banks being subject to slip. But no part of the outer slope should fall within a line drawn from the water's edge at an angle of 1 in 2½. A bank of greater weight is better, because more stable, and the mere weight conduces largely to a perfectly water-tight condition. The outer slopes should be sown with short grass. Sheep grazed on the banks improve the surface by treading, but large cattle may cause injury by collection of water in puddles.

Use of two ponds.—In most cases two ponds are used, so that the hot water may be first run into one and cool partially, after which it is run into the other one, from which the engine supply is taken. By this means cooler water is secured than is possible with a single pond. The bank between the two should be treated with all the care of an outer bank. The water should be allowed to run equally gently into each pond, and from each should be withdrawn in the same manner from below the surface. Each pond should be provided with a separate drain-pipe. Generally, also, the two ponds should be arranged throughout so that either one may be used separately at any time, while the other is emptied for any reason.

Area of pond.—The area of pond required depends primarily upon the amount of heat which is required to be dispersed in the particular case. A very close estimate of the amount of this heat may be based upon the consumption of coal. The quantity of heat expended in the furnace per hour is to be first ascertained, by multiplying the number of pounds of coal burnt by its calorific power, which for good average coal is about 13,000 units. In best practice, three-fourths of this is expended in actual evaporation of water, and the quantity of heat thus determined is assumed to be supplied to the engine. The mechanical work performed by the engine causes the disappearance of 42·75 units of heat per minute, or 2565 units per hour per indicated horse-power. All the balance of the heat may be regarded as passing from the engine, through the condenser and air-pump, to the cooling-pond. In fair weather each square foot of water-surface, in a cooling-pond with a good exposure, may be relied upon to effect the dispersion of 600 units of heat per hour. The area in square feet of pond required in any particular case therefore equals—

$$\frac{(a \times b \times c) - (d \times e)}{f}$$

a = Pounds of coal consumed per hour.
b = Calorific power of coal = say 13,000.
c = Co-efficient or proportion of heat utilized = say ·75.
d = Thermal units of heat per horse-power per hour which disappear in the production of mechanical work = 2565.
e = Indicated horse-power.
f = Units of heat dissipated per square foot of water-surface per hour.

Applying the above to a case in which 2000 pounds of coal are burnt per hour to produce 1000 indicated horse-power, it is found that an area of 28,225 square feet, or ·65 acre of water-surface, is required in the pond. The conclusions thus arrived at are fairly approximate, but if the results of actual trials upon the same or a similar engine are available from which the amount of rejected heat can be deduced, this should be adopted, for the sake of simplicity and greater accuracy. The area in square feet will then be obtained by dividing by 600 the number of units per hour dispersed per sq. ft. The greatest difference is likely to arise in connection with the boiler, whose efficiency has been taken rather high. If, however, the boiler is less efficient, the co-efficient c should be reduced. Good boilers, in every way perfect of their kind, often show a co-efficient of ·60, which would bring the area of cooling-pond one-fifth less than the assumed co-efficient of ·75, simply because no cooling area is required on account of heat sent up the chimney. One frequent reason for this is the fact that in such cases the boilers are very heavily worked. There are also various small losses of heat about the engine which, if carefully measured, would

lead to a trifling reduction in the necessary area of pond. The area is, however, obtained with sufficient accuracy for ordinary cases by the means described, on the assumption that work is suspended by night. At all times the temperature of the water in the pond must rise somewhat, by reason of the heat poured into it from the engine. The rate of surface-cooling increases as the temperature rises, so that in ordinary weather it soon reaches 600 units per square foot, when, if the pond is accurately proportioned, its temperature fluctuates in approximate correspondence to the variations in the temperature of the atmosphere.

In calm, damp, warm weather, the dispersion of heat falls to one-half of its normal rate—more or less, according to the character of the surroundings. This causes an increase in the temperature of the pond, and a condition of equilibrium may not be reached until the water becomes too hot for use. When work ceases during the night, the surface-cooling of the pond continues or probably increases, owing to the lower temperature of the atmosphere. If half of the quantity of heat should be allowed to accumulate in the water during a period of 10 hours, the temperature of the upper two feet in depth will be raised by an average of 24° F. Supposing the cooling power to remain constant during the night and day, there would be but little more than sufficient time to cool the water before morning. If work is required to be continued through the night and day, the area of pond provided should be almost, if not quite, twice as great as obtained by the above method.

If the conditions affecting a cooling-pond are exceptionally favourable, a corresponding reduction may be made in its area. Assuming, however, that a reduction of one-fourth is made in the area obtained as above,

and that during unfavourable weather it is found to be impossible to effect a dispersion of more than 300 thermal units per square foot per hour, the average temperature of the upper two feet in depth will be increased by 40° in 10 hours. In a similar way, if the area is reduced to one-half, the upper three feet in depth will be raised in temperature 48° by 10 hours' work. It is therefore impossible to recommend any reduction in the area as obtained by rule, unless accompanied by exceptionally favourable conditions, or unless the cooling-pond is relieved by other appliances.

Depth of pond.—The depth of a pond may be four to five feet. A pond of greater depth is convenient, in case night cooling is largely depended upon, as it gives a larger volume of water to receive the accumulated heat. But it is apt to be neglected, so that its depth becomes reduced by accumulated mud.

Storage in a separate pond.—In case the storage of water should be considered to be necessary in anticipation of drought or irregularity in supply, it is better to provide a separate pond, rather than to deepen the cooling-pond with such object. A storage-pond is all the better if made of ample depth, and of comparatively small surface area, so as to reduce the loss of water by evaporation in dry weather. A separate pond full of water can be kept ready for filling up the cooling-pond after cleaning out; and this convenience will conduce to more frequent cleaning. The use of a cooling-pond for storage purposes necessitates so much variation in the level of the water-surface, that its value for cooling purposes is much impaired. The loss of water by surface evaporation is about four inches per month in April and May, when dry winds prevail. In June to September it reaches nearly two inches per month. These figures apply to cases in which the temperature

of the water is approximately equal to that of the atmosphere. When the temperature of the water exceeds that of the atmosphere, the above rates of evaporation are exceeded. The dispersion of 600 units of heat per square foot by pure evaporation would reduce the level of the water-surface by ·12 inch.

Auxiliary cooling.—Various arrangements are adopted to secure a preliminary partial cooling of the water before it is allowed to mix in the pond. These are: (1) long low troughs acting by surface cooling, while the water is in an agitated condition, (2) high troughs perforated to distribute the water in a fine shower, (3) cascade with small steps, (4) cooling-stacks, (5) centrifugal distributors. The whole of these arrangements act by promoting the intermixture or contact of water in motion with air. Under such conditions, the transfer of heat from the water to the air is greater than when both are still. More evaporation also takes place, so that more heat is abstracted from the unevaporated water, and rendered latent in the vapour. When the water is allowed to fall freely from a height, air is found to be dissolved to a greater extent than when a still surface of water is exposed to the atmosphere. This air is again liberated when the water re-enters the condenser. If the air-pump is of the highest degree of efficiency, the air is rapidly withdrawn, but otherwise the presence of unnecessary air in the condenser causes a serious loss in vacuum. All such means should therefore be avoided in every possible case.

Cooling-stacks.—Cooling-stacks or towers are used either as auxiliaries to assist cooling-ponds, or as substitutes to entirely replace ponds. They are composed of boards, slates, cloths, brushwood, or other material, so disposed as to expose a large surface, over which the water used for condensation is allowed to trickle and

to become cooled by exposure to the atmosphere. In some cases the water is allowed a clear drop of considerable depth, or a smaller drop repeated several times. In either case, more air is absorbed by the water than when it is allowed to trickle in contact with a surface. Generally the water is required to be raised by pumping, which may be effected either by the use of a closed air-pump, a separate reciprocating-pump (for which purpose the Worthington type is exceedingly well adapted), or a centrifugal-pump. Great care should be taken to ensure uniformity of distribution of the water over the whole of the surfaces, or that any departure from strict uniformity should be in the direction of increased supply to such surfaces as enjoy the best exposure to the air. A fan is in many cases employed to ensure intimate contact between the air and the water to be cooled. A comparatively small amount of heat is dispersed by radiation. In the absence of mechanical appliances for producing a brisk current of air, but little heat is dispersed by direct conduction to the air, in which case practically the whole of the heat is dispersed by evaporation. By the use of a fan the cooling power of the stack is very largely increased. When this course is adopted a large quantity of heat is carried away by reason of the increased temperature of the air. But much water is carried off in the form of a cloud of mist or imperfectly-evaporated water, so that it is found in practice that the loss of water is almost precisely equal, in ordinary weather, to the amount supplied to the boiler; but in very cold weather, when the air is supplied at a low temperature, the loss is rather less. The number of units of heat dispersed per square foot of wetted surface depends upon the quantity of air supplied, and upon the amount by which its temperature is increased.

The latter factor depends upon the direction of the surfaces as compared with the current, upon the length of wetted surface traversed by the air, upon the temperature of the air, and that of the water. Obviously, the circuits should be so arranged that the coolest water will be exposed to the coolest air. It is stated that under favourable conditions, 4000 units of heat have been dispersed per hour per square foot of wetted surface, but except for the most urgent reasons, this should not be allowed to exceed 600 to 1000 units per square foot per hour. Adopting 600 units as the rate of cooling, the apparatus may be so designed that the total area of ground occupied is only about one per cent. of that necessary for an ordinary cooling-pond. The use of a fan renders the apparatus practically independent of the natural state of the atmosphere as to wind or stagnation. Some advantage still remains in favour of a dry atmosphere, but provision should be made for driving the fan at an increased speed during the prevalence of damp weather. The concentration of the operation of heat dispersion naturally leads to a local intensification of the heated, foggy condition of the air. A chimney or shaft should therefore be adopted to carry off the vapour beyond the possibility of offence. This, however, need not be nearly so high as one for carrying off smoke. When the cooled water has not been allowed to fall considerable distances so as to become aerated, it is found to be remarkably free from dissolved air, and that the vacuum in the condenser is exceptionally good. The use of a cooling-stack is attended by the necessity for some expenditure of power for driving the pump and the fan, but with judicious proportions this will be small.

Partial disposal of impurity and heat in over-flow water.—When the water-supply is slightly in excess

of all requirements for boiler feeding—allowing for ample blowing-off—and for the supply of the pond or tower, some will be allowed to run to waste. When such is the case, and no objection is likely to arise from outside, a great advantage will be secured by running the over-flow from the air-pump into a tank, and passing all excess water from this tank. Such waste should be collected from the surface, so as to dispose of any floating oil or scum, and also because by this means the water may be sent away at a slightly higher temperature, and a cooler supply secured.

Pond or stack at low level.—When the vacuum is formed in a condenser, the injection water may be drawn from a pond whose level is below the injection-valve by an amount which varies with the perfection of the vacuum, but which in almost every case may reach 18 or 20 feet without introducing any difficulty. When the pond is nearly at the level of the condenser, the injection-valve must be partially closed to interpose a resistance, and prevent the entry of an excess of water. By this means the temperature of the injection water is increased by an exceedingly small amount, so that in theory the lower pond would be really the most efficient. In some cases, in which the pond is at a low level, the vacuum is not so easily established on first starting the engine. But a pond at a low level may be often constructed with greater convenience than one at a high level. Similarly also, when a cooling-stack is adopted, the pumping may be materially relieved by a judicious arrangement of levels.

CHAPTER XVII.

HEAT-RETAINING COVERINGS.

Transfer of heat.—When two bodies of different temperatures are in contact, the one whose temperature is highest imparts heat to the other one, so that after a time the two assume an equal temperature. The two in combination may gain heat from, or lose heat to, other bodies; but as concerns each other, one will lose exactly as many units of heat as are gained by the other. This transfer may in the first instance be regarded as taking place entirely by conduction, aided by circulation or convection, when one or both bodies exist in a fluid condition. Radiation occurs only upon the surface, and will be subsequently considered. Heat is transferred most freely by bodies of high conductivity. Conversely, bodies of low conductivity are employed to interpose resistance to such transfer of heat; and substances suitable for this purpose have become known as "non-conductors," which term is sufficiently expressive, though not strictly correct.

The value of steam is for most purposes very much dependent upon its pressure, and this suffers great reduction upon the loss of a comparatively small proportion of the total amount of heat contained within

it. The protection of steam from loss of its contained heat is therefore a matter of great importance, as bearing upon economical working. The same treatment which is necessary for successful use of steam is equally appropriate for other purposes, of which perhaps the most notable is that of the protection of objects which have been artificially cooled, so as to prevent them from absorbing heat from surrounding bodies.

Transfer of heat impeded by substances of low conductivity.—The most obvious means for preventing loss of heat by steam is to confine it in a vessel made of material of low conducting power, if such can be obtained. But practically metals are necessary for the construction of vessels of strength sufficient to withstand steam-pressures. All metals are good conductors of heat, though they vary considerably when compared with each other. A coating of some substance of low conducting power is necessary for application to the outside of a metal vessel, in order to obstruct the wasteful passage of heat from the steam inside the vessel to the atmospheric air outside the vessel.

Circulation of fluid, or convection, to be prevented.—Air is a bad conductor of heat, but effects transfer by convection when the conditions are favourable. The simplest covering for retaining heat consists of a closed jacket or space filled with air, which is prevented from circulating or changing. Probably the most favourable condition in this connection is found when the hot surface is flat and horizontal, and the air-jacket is below it, so that any heat which is imparted to it is quite unable to give rise to a circulation of air. If the jacket is above the heated surface, the circulation which is produced effects a rapid transfer of heat from

the inner to the outer part of the jacket, which is thus of comparatively little use as a heat-retainer. In the same way vertical air-jackets are of little use when the contained air is allowed to move.

Coverings in which circulation of air is retarded by fineness of division.—When the circulation of air is placed under efficient restraint, the jacket at once becomes valuable. In some cases this may be effected by the use of diaphragms, or structural means whereby the circulation is localized. But usually the restraint is much more effective when applied in the form of a finely-divided substance which fills the space, and opposes the passage of heat by presenting an infinite number of minute particles through which the heat must pass, alternatively with the interposed atoms of air, before it can become dissipated in the outer atmosphere. The efficiency of this arrangement is in proportion to the insulating power of the substance, and to the fineness of its division. As a general rule, substances which contain most air are the best non-conductors, but subject to modification according to the perfection with which the contained air is prevented from moving. Feathers, hair, cotton, coir, and other fibres of organic origin, are more efficient than any other substances. Cork powder and fine sawdust are next, followed by mineral fibres and infusorial earth. Almost all minerals, reduced to powder, oppose some resistance to the passage of heat, by reason of the contained air, but this is imperfectly restrained from moving, and as a rule the solid substance possesses a high degree of conductivity, so that the ultimate efficiency is low. In comparing substances of approximately similar composition, preference should be accorded to those of lowest specific gravity,—other

conditions being equal,—as these contain more air to oppose resistance and less solid matter to transmit heat.

Durability of coverings.—In addition to the power to resist the passage of heat, a covering is practically required to possess strength to resist abrasion, and chemical stability to resist destructive decomposition. As a rule, the best insulating materials used alone are quite unable to oppose the requisite degree of resistance under wear. This is, however, imparted by the use of an outer covering of canvas, wire-netting, hoop-iron,—interwoven, after the manner of basket-work,—close boarding secured by hoop-iron, boarding with various degrees of spacing, also secured by hoop-iron, and sheet-iron, secured in various ways. When in perfect condition, organic substances are undoubtedly the most perfect insulators. But in every case they are prone to develop slow destructive decomposition, which generally transforms them to some kind of carbonaceous powder. If this is absolutely prevented from displacement, it still possesses a considerable insulating power, though less than possessed by the substance in its perfect condition. In the process of decomposition, however, a shrinkage takes place, the material becomes loose, so that displacement is facilitated, and portions of the surface are liable to be left without protection. It is therefore found that this class of material is of comparatively little value for direct application to very hot surfaces, except for clothing horizontal steam-pipes of moderate size, where the bulk of the material is insufficient to lead to displacement. In such cases it is protected by a parcelling of canvas, painted over. For vertical pipes, especially if exposed to vibration, it is distinctly unsuitable. Clay and other plastic materials are used very

largely in connection with organic fibres, with a view to prevent the change of form which often follows decomposition, and also to facilitate the application to the surfaces. The extremely low cost of clay often leads to its use in excessive proportions. The hair or other fibre should be liberally supplied, and also some material which will induce a porous condition in the mass when finished. For the last purpose, material which shrinks upon drying but does not cause cracking should be used in fair proportion, for which purpose cow-dung is largely adopted. This also is a cheap material, and therefore apt to displace the fibre which forms the really effective coating.

Comparative permanence of mineral coverings.—With a view to avoid the decomposition to which all substances of organic origin are subject, various substances of mineral origin are used, of which the most important are asbestos and infusorial earth or *kieselguhr*. These are practically unaffected by decomposition, and are rather better adapted to resist abrasion than are the organic compositions. They are decidedly less efficient insulators while the latter are in good condition. A good average coating of these compositions is, however, much superior to one of organic composition which has been allowed to deteriorate in the usual manner.

Application of compositions.—As a rule, both organic and mineral compositions are applied in the form of a mortar, generally to the hot surface. If a continuous coat of moderate thickness were to be applied, the shrinkage upon drying would cause the whole to fall off in large pieces. To prevent this, the first application is made either in a thin film or wash well rubbed on, or otherwise in small dabs. This must be allowed to dry thoroughly, and then followed by

successive thin coats, each of which is allowed to dry very slowly and complete its shrinking before the application of the succeeding one.

The original surface to which the coating is applied must be cleaned quite free from rust, grease, dirt, or loose matter of any kind. A trowel is used in the application, and the last coat is rubbed quite smooth when partially dried. A covering hastily applied in great thicknesses may hold together and superficially appear to be perfect. But it is apt to separate bodily from the plates, whereby the object of its application is partially defeated, any leakage which may become developed is likely to remain concealed, and corrosion is promoted.

Slag wool.—Slag wool or silicate cotton is used by dry packing in any space available, and covering by any of the means used for the protection of compositions. This is an efficient covering while in good condition, but a high temperature, acting simultaneously with pressure, causes it to shrink into a dense glassy condition, in which it is practically useless for the required purpose. When dry, and subject to concussion or vibration, it wastes and gives off a dust which is prejudicial to sliding surfaces or bearings, and therefore it should not be used in close proximity to moving machinery, especially on shipboard. When subjected to the action of moisture, the small amount of sulphur which it contains is prone to originate a corrosive action upon ironwork, of a nature very similar to that produced by furnace ashes.

Felt.—Hair felt of different thicknesses is largely used for covering pipes, lining doors, and for other similar work. This is highly efficient but costly, and far from durable if exposed to high temperatures or

rough wear. Probably the best covering which can be applied at the present time is one in which a coating of mineral composition is first applied to the hot surface, a thick covering of felt applied outside this, and the whole protected from wear and the weather.

Removable coverings.—Coverings of various classes have been recently made in blocks, slabs, or ribbons ready for application to the surfaces. These are very convenient, and possess an important advantage in their less liability to conceal small leakages which may exist unsuspected beneath a solid coating. They may also be removed at any time for purposes of cleaning, examination, and repair to any part, and may afterwards be replaced without damage. But they are very costly, and are liable to admit access of cold air unless the fitting is excellent. Slag wool is used in this manner, sewn between two thicknesses of wire-netting, at a moderate cost. The French Admiralty also use asbestos cloth with asbestos fibre placed between, the whole being sewn with asbestos thread, to a thickness of 1 to 1½ inches. This may be cut to any required shape. Papier-maché blocks moulded to special shapes have also been used.

Small portions of an insulating covering have been long adopted in doors or hatches, of boarding lined with thick hair felt, arranged to remove for convenient access to special parts, and secured by screws. The felt should be secured to the wood by means of galvanized wire-netting with a close mesh, or by battens, particular attention being directed towards security at the edges, and to good fitting generally.

Covering of long steam-pipes.—In some cases steam-pipes are required to be carried long distances, when very good protection is secured by wood boxes, filled

with dry sawdust or charcoal, not less than from two to three inches thick in any part. In such cases damp causes decomposition of sawdust, and increases the conducting power of all substances, or impairs their insulating power. Sawdust or charcoal will absorb a large quantity of moisture arising from pipe leakage, and prevent its discovery at an early stage. It is therefore particularly important that the pipes should be thoroughly well jointed before covering, and that facility for free expansion should be provided. Obviously, when pipes, coated or uncoated, are in the open air, some covering should be provided to protect them from rain, snow, or drip of any kind. In 1865 a 4-inch steam-pipe at the Gould and Curry mine was carried most successfully a distance of 1300 feet underground in boxes eight inches square inside and two inches thick, the pipe being surrounded with wood ashes.

Boxes containing insulated steam-pipes may be buried in the ground, provided they be protected from moisture from above and from below, the latter being often the more important. Furnace ashes produced from coal should never be used in any way for such a purpose, as they exercise a very serious corrosive influence upon all ironwork, and possess practically no insulating power. Mr. Leavitt has used a mixture of 1 plaster-of-Paris to 2 of sawdust, applied as mortar 2½ inches thick, and found that when the composition has become dry the condensation is only about one-hundredth part of that previously observed in the bare pipes.

Radiation from surfaces.—The character of the outer surface of a covering which is exposed to the air affects its radiating power, and consequently the amount of heat lost. A bright polished surface radiates less heat than a dull one. A surface of dark colour generally

radiates more heat than one of a light colour. Surfaces which give off heat freely by radiation also possess a correspondingly high degree of absorptive power for heat which is radiated from a higher source, but this fact is of very limited application in the present connection. The loss by radiation is reduced by painting the exposed surfaces in good smooth oil paint of light colour. This has a further advantage in many cases, by reason of protection to the surface against wear and weather.

External radiating surfaces to be minimized.—In connection with the question of loss by heat radiation, the quantity of surface exposed is an important factor. Steam-pipes and other details are often made unnecessarily large. Though certain advantages are secured in this way, better economical results would be obtained by the use of less ample proportions.

Treatment of internal radiating surfaces.—Professor Thurston recently proposed to treat the unpolished or non-frictional surfaces of an engine with a view to impede the transfer of heat. These surfaces were treated with dilute sulphuric acid for forty-eight hours, so as to cause disintegration of the surface, after which linseed oil or varnish was applied in one or more coats. This treatment is said to cause a reduction of 40 per cent. in the transmission of heat. The immediate object of the proposal is the reduction of cylinder condensation, but if successful for this purpose there is no reason why it should not be equally so for other work.

Protection against access of heat.—The means of insulation which have been described are generally adopted with the object of protecting bodies or substances whose temperatures have been raised by artificial means above that of the surrounding atmosphere,

and of preventing them from losing heat, so as to cause a reduction in temperature. But operations are now largely resorted to for the purpose of reduction of temperature; and it is equally important that the work be protected from the access of heat from any external source, which would vitiate the result equally with the loss of heat in an ordinary case. In such cases practically the same means are available, with the advantage that chemical decomposition is much less active than it is at high temperature. It is, however, of even greater importance to prevent the access of damp to the insulating medium. The low temperature causes the condensation of water-vapour, and the accumulation of the water produced. This seriously impairs the insulation at once. But if the temperature is sufficiently low—as it usually is—the whole of the insulating coating becomes permeated with ice, when its value is destroyed. The prejudicial effect of air in the coating and in the cold room is often greater than is suspected. Vertical doors in the walls of such a room admit large amounts of air at ordinary temperatures, which causes loss of cooling effect. This is much reduced by providing access to the cold room by means of a scuttle or hatchway in the ceiling of the room. Charcoal of small size, produced by the charring of wood shavings and machine refuse, forms an efficient covering when exposed to low temperatures, but powdered charcoal should not be used, on account of its liability to spontaneous combustion. Charcoal produced from the branches and twigs of trees contains too much air in the spaces between the pieces. It is therefore comparatively useless, unless it be crushed sufficiently to pack into a more solid condition. Kiln-dried sawdust, slag wool, paper, and boarding are much more

largely used in this work than is practicable in hot work.

Protection against frost.—The principles which govern the protection of hot bodies against cooling by conduction and radiation also apply to the treatment of water-pipes, pumps, and other vessels for protection against frost. Also generally to all cases in which the passage of heat is required to be obstructed, provided that the temperature and chemical conditions encountered are not such as to cause the destructive decomposition of the insulating medium selected.

TABLE IX.

THE PROPERTIES OF SATURATED STEAM AT EACH DEGREE OF TEMPERATURE, FROM 32° TO 213·3° FAHRENHEIT, OR FROM ·089 OF A POUND TO 15 POUNDS PRESSURE ON THE SQUARE INCH.

Temperature of the vapour, and of the water from which it is evaporated.	Number of British Thermal Units contained in one pound, reckoned from zero Fahrenheit.				Elastic Force; expressed in:		Weight of one cubic foot, in decimals of a pound.
	Number contained in water.	Number required for evaporation, known as latent heat, or heat of vaporization.	Total number contained in the vapour.	Difference.	Pounds on the square inch.	Inches of mercury at 32°.	
a	b	c	d	e	f	g	h
32°	32·000	1091·700	1123·700	·305	·089	·1811	·00030
33	33·000	1091·005	1124·005	·305	·092	·1884	·00030
34	34·000	1090·310	1124·310	·305	·096	·1960	·00031
35	35·000	1089·615	1124·615	·305	·100	·2039	·00032
36	36·000	1088·920	1124·920	·305	·104	·2121	·00033
37	37·000	1088·225	1125·225	·305	·108	·2205	·00034
38	38·000	1087·530	1125·530	·305	·112	·2292	·00036
39	39·001	1086·834	1125·835	·305	·117	·2382	·00038

40	40·001	1086·139	1126·140	·305	·122	·2476	·00040
41	41·001	1085·444	1126·445	·305	·127	·2573	·00042
42	42·001	1084·749	1126·750	·305	·132	·2673	·00043
43	43·001	1084·054	1127·055	·305	·137	·2777	·00045
44	44·002	1083·358	1127·360	·305	·142	·2884	·00047
45	45·002	1082·663	1127·665	·305	·147	·2994	·00049
46	46·002	1081·968	1127·970	·305	·152	·3109	·00050
47	47·002	1081·273	1128·275	·305	·158	·3228	·00052
48	48·003	1080·577	1128·580	·305	·164	·3351	·00054
49	49·003	1079·882	1128·885	·305	·170	·3478	·00056
50	50·003	1079·187	1129·190	·305	·176	·3608	·00058
51	51·004	1078·491	1129·495	·305	·183	·3743	·00060
52	52·004	1077·796	1129·800	·305	·190	·3883	·00062
53	53·005	1077·100	1130·105	·305	·197	·4028	·00065
54	54·005	1076·405	1130·410	·305	·205	·4177	·00067
55	55·006	1075·709	1130·715	·305	·212	·4332	·00069
56	56·006	1075·014	1131·020	·305	·220	·4492	·00071
57°	57·007	1074·318	1131·325	·305	·228	·4656	·00073

a	b	c	d	e	f	g	h
58°	58·007	1073·623	1131·630	·305	·236	·4825	·00076
59	59·008	1072·927	1131·935	·305	·245	·5000	·00079
60	60·009	1072·231	1132·240	·305	·254	·5180	·00082
61	61·010	1071·535	1132·545	·305	·263	·5367	·00085
62	62·011	1070·839	1132·850	·305	·273	·5560	·00088
63	63·012	1070·143	1133·155	·305	·282	·5758	·00091
64	64·013	1069·447	1133·460	·305	·292	·5962	·00094
65	65·014	1068·751	1133·765	·305	·302	·6173	·00097
66	66·015	1068·055	1134·070	·305	·313	·6391	·00100
67	67·016	1067·359	1134·375	·305	·324	·6615	·00103
68	68·018	1066·662	1134·680	·305	·335	·6846	·00107
69	69·019	1065·966	1134·985	·305	·347	·7084	·00111
70	70·020	1065·270	1135·290	·305	·359	·7330	·00115
71	71·021	1064·574	1135·595	·305	·372	·7583	·00119
72	72·023	1063·877	1135·900	·305	·385	·7844	·00123
73	73·024	1063·181	1136·205	·305	·398	·8114	·00127
74	74·026	1062·484	1136·510	·305	·411	·8391	·00131
75	75·027	1061·788	1136·815	·305	·425	·8676	·00135

76	76·029	1061·091	1137·120	·305	·440	·8969	·00139
77	77·030	1060·395	1137·425	·305	·455	·9271	·00143
78	78·032	1059·698	1137·730	·305	·470	·9583	·00148
79	79·034	1059·001	1138·035	·305	·486	·9905	·00153
80	80·036	1058·304	1138·340	·305	·502	1·023	·00158
81	81·037	1057·608	1138·645	·305	·518	1·056	·00163
82	82·039	1056·911	1138·950	·305	·535	1·091	·00168
83	83·041	1056·214	1139·255	·305	·553	1·127	·00173
84	84·043	1055·517	1139·560	·305	·571	1·163	·00178
85	85·045	1054·820	1139·865	·305	·590	1·201	·00183
86	86·047	1054·123	1140·170	·305	·609	1·240	·00189
87	87·049	1053·426	1140·475	·305	·629	1·281	·00195
88	88·051	1052·729	1140·780	·305	·650	1·323	·00201
89	89·053	1052·032	1141·085	·305	·671	1·366	·00207
90	90·055	1051·335	1141·390	·305	·692	1·410	·00213
91	91·057	1050·638	1141·695	·305	·715	1·454	·00219
92	92·059	1049·941	1142·000	·305	·738	1·500	·00226
93°	93·061	1049·244	1142·305	·305	·761	1·548	·00233

a	b	c	d	e	f	g	h
94°	94·063	1048·547	1142·610	·305	·785	1·597	·00240
95	95·065	1047·850	1142·915	·305	·809	1·647	·00247
96	96·068	1047·152	1143·220	·305	·834	1·698	·00254
97	97·071	1046·454	1143·525	·305	·860	1·751	·00262
98	98·074	1045·756	1143·830	·305	·887	1·805	·00270
99	99·077	1045·058	1144·135	·305	·914	1·861	·00278
100	100·080	1044·360	1144·440	·305	·943	1·918	·00286
101	101·083	1043·662	1144·745	·305	·972	1·977	·00294
102	102·086	1042·964	1145·050	·305	1·001	2·037	·00302
103	103·089	1042·266	1145·355	·305	1·031	2·099	·00311
104	104·092	1041·568	1145·660	·305	1·062	2·163	·00320
105	105·095	1040·870	1145·965	·305	1·094	2·227	·00330
106	106·098	1040·172	1146·270	·305	1·126	2·293	·00340
107	107·101	1039·474	1146·575	·305	1·159	2·361	·00350
108	108·104	1038·776	1146·880	·305	1·193	2·431	·00360
109	109·107	1038·078	1147·185	·305	1·229	2·503	·00370
110	110·110	1037·380	1147·490	·305	1·265	2·577	·00380
111	111·113	1036·682	1147·795	·305	1·302	2·653	·00390
112	112·117	1035·983	1148·100	·305	1·341	2·731	·00400

113	113·121	1035·284	1148·405	·305	1·381	2·810	·00410
114	114·125	1034·585	1148·710	·305	1·421	2·892	·00421
115	115·129	1033·886	1149·015	·305	1·462	2·976	·00433
116	116·133	1033·187	1149·320	·305	1·504	3·061	·00445
117	117·137	1032·488	1149·625	·305	1·547	3·149	·00457
118	118·141	1031·789	1149·930	·305	1·591	3·239	·00470
119	119·145	1031·090	1150·235	·305	1·636	3·331	·00483
120	120·149	1030·391	1150·540	·305	1·682	3·425	·00496
121	121·153	1029·692	1150·845	·305	1·730	3·522	·00508
122	122·157	1028·993	1151·150	·305	1·779	3·621	·00521
123	123·161	1028·294	1151·455	·305	1·828	3·723	·00535
124	124·165	1027·595	1151·760	·305	1·879	3·826	·00549
125	125·169	1026·896	1152·065	·305	1·931	3·933	·00563
126	126·173	1026·197	1152·370	·305	1·984	4·042	·00578
127	127·177	1025·498	1152·675	·305	2·039	4·153	·00593
128	128·182	1024·798	1152·980	·305	2·096	4·267	·00608
129	129·187	1024·098	1153·285	·305	2·154	4·384	·00624
130°	130·192	1023·398	1153·590	·305	2·213	4·503	·00640

a	b	c	d	e	f	g	h
131°	131·197	1022·698	1153·895	·305	2·273	4·625	·00656
132	132·202	1021·998	1154·200	·305	2·335	4·750	·00673
133	133·207	1021·298	1154·505	·305	2·398	4·878	·00690
134	134·212	1020·598	1154·810	·305	2·461	5·009	·00707
135	135·217	1019·898	1155·115	·305	2·526	5·143	·00725
136	136·222	1019·198	1155·420	·305	2·594	5·280	·00743
137	137·227	1018·498	1155·725	·305	2·663	5·420	·00761
138	138·233	1017·797	1156·030	·305	2·732	5·563	·00780
139	139·239	1017·096	1156·335	·305	2·803	5·709	·00799
140	140·245	1016·395	1156·640	·305	2·876	5·858	·00819
141	141·251	1015·694	1156·945	·305	2·952	6·011	·00839
142	142·257	1014·993	1157·250	·305	3·029	6·167	·00860
143	143·263	1014·292	1157·555	·305	3·108	6·327	·00881
144	144·269	1013·591	1157·860	·305	3·188	6·490	·00903
145	145·275	1012·890	1158·165	·305	3·270	6·657	·00925
146	146·281	1012·189	1158·470	·305	3·353	6·827	·00948
147	147·287	1011·488	1158·775	·305	3·438	7·001	·00971
148	148·293	1010·787	1159·080	·305	3·526	7·179	·00993

149	149·299	1010·086	1159·385	·305	3·615	7·361	·01016
150	150·305	1009·385	1159·690	·305	3·707	7·547	·01040
151	151·311	1008·684	1159·995	·305	3·801	7·736	·01064
152	152·318	1007·982	1160·300	·305	3·896	7·929	·01089
153	153·325	1007·280	1160·605	·305	3·992	8·127	·01114
154	154·332	1006·578	1160·910	·305	4·090	8·329	·01140
155	155·339	1005·876	1161·215	·305	4·191	8·535	·01167
156	156·346	1005·174	1161·520	·305	4·295	8·745	·01194
157	157·353	1004·472	1161·825	·305	4·400	8·959	·01222
158	158·360	1003·770	1162·130	·305	4·507	9·178	·01250
159	159·367	1003·068	1162·435	·305	4·617	9·401	·01279
160	160·374	1002·366	1162·740	·305	4·729	9·629	·01308
161	161·381	1001·664	1163·045	·305	4·843	9·861	·01338
162	162·389	1000·961	1163·350	·305	4·960	10·098	·01368
163	163·397	1000·258	1163·655	·305	5·079	10·340	·01399
164	164·405	999·555	1163·960	·305	5·200	10·588	·01430
165	165·413	998·852	1164·265	·305	5·324	10·840	·01462
166°	166·421	998·149	1164·570	·305	5·451	11·097	·01495

a	b	c	d	e	f	g	h
167°	167·429	997·446	1164·875	·305	5·580	11·359	·01528
168	168·437	996·743	1165·180	·305	5·711	11·627	·01562
169	169·445	996·040	1165·485	·305	5·845	11·900	·01596
170	170·453	995·337	1165·790	·305	5·981	12·178	·01631
171	171·461	994·634	1166·095	·305	6·120	12·461	·01667
172	172·470	993·930	1166·400	·305	6·262	12·750	·01704
173	173·479	993·226	1166·705	·305	6·408	13·045	·01741
174	174·488	992·522	1167·010	·305	6·555	13·345	·01779
175	175·497	991·818	1167·315	·305	6·704	13·651	·01817
176	176·506	991·114	1167·620	·305	6·857	13·963	·01855
177	177·515	990·410	1167·925	·305	7·013	14·281	·01894
178	178·524	989·706	1168·230	·305	7·172	14·605	·01934
179	179·533	989·002	1168·535	·305	7·335	14·935	·01975
180	180·542	988·298	1168·840	·305	7·500	15·271	·02017
181	181·551	987·594	1169·145	·305	7·668	15·614	·02060
182	182·561	986·889	1169·450	·305	7·841	15·963	·02104
183	183·571	986·184	1169·755	·305	8·016	16·318	·02148
184	184·581	985·479	1170·060	·305	8·194	16·680	·02193

185	185·591	984·774	1170·365	·305	8·375	17·049	·02238
186	186·601	984·069	1170·670	·305	8·558	17·425	·02284
187	187·611	983·364	1170·975	·305	8·745	17·807	·02331
188	188·621	982·659	1171·280	·305	8·936	18·196	·02379
189	189·632	981·953	1171·585	·305	9·132	18·593	·02428
190	190·643	981·247	1171·890	·305	9·330	18·997	·02470
191	191·654	980·541	1172·195	·305	9·532	19·408	·02529
192	192·665	979·835	1172·500	·305	9·738	19·827	·02580
193	193·676	979·129	1172·805	·305	9·947	20·253	·02632
194	194·686	978·424	1173·110	·305	10·160	20·687	·02685
195	195·697	977·718	1173·415	·305	10·377	21·129	·02740
196	196·708	977·012	1173·720	·305	10·597	21·579	·02796
197	197·719	976·306	1174·025	·305	10·822	22·036	·02853
198	198·730	975·600	1174·330	·305	11·051	22·502	·02910
199	199·741	974·894	1174·635	·305	11·284	22·976	·02967
200	200·753	974·187	1174·940	·305	11·521	23·458	·03025
201	201·765	973·480	1175·245	·305	11·761	23·948	·03083
202°	202·777	972·773	1175·550	·305	12·006	24·446	·03142

a	b	c	d	e	f	g	h
203°	203·789	972·066	1175·855	·305	12·255	24·953	·03201
204	204·801	971·359	1176·160	·305	12·508	25·468	·03261
205	205·813	970·652	1176·465	·305	12·766	25·992	·03323
206	206·825	969·945	1176·770	·305	13·028	26·525	·03386
207	207·837	969·238	1177·075	·305	13·295	27·067	·03450
208	208·849	968·531	1177·380	·305	13·568	27·619	·03516
209	209·861	967·824	1177·685	·305	13·843	28·180	·03584
210	210·874	967·116	1177·990	·305	14·122	28·751	·03654
211	211·887	966·408	1178·295	·305	14·406	29·332	·03725
212	212·900	965·700	1178·600	·305	14·696	29·9218	·03797
213	213·913	964·992	1178·905	·008	14·991	30·522	·03871
213·03°	213·939	964·974	1178·913		15·000	30·540	·03873

TABLE X.

The Properties of Saturated Steam, at each Pound of Pressure, from One Pound to Two Hundred and Sixty Pounds on the Square Inch.

Elastic Force.		Temperature in degrees Fahrenheit of the steam, and of the water from which it is evaporated.	Number of British Thermal Units contained in one pound, reckoned from zero Fahrenheit.						Weight of one cubic foot, in decimals of a pound.	Gauge pressure, in lbs. on the square inch, with barometer at 29·922 inches.
In lbs. on the square inch.	In inches of mercury at 32°.		Number contained in the water.	Difference.	Number required for evaporation, known as latent heat, or heat of vaporization.	Total number contained in the steam.	Difference.	2nd Difference.		
p	q	r	s	t	u	v	w	x	y	z
1	2·036	102·	102·086	24·354	1042·964	1145·050	7·400		·0030	..
2	4·072	126·266	126·440	15·437	1026·010	1152·450	4·681	2719	·0058	..
3	6·108	141·622	141·877	11·519	1015·254	1157·131	3·494	1187	·0085	..
4	8·144	153·070	153·396	9·326	1007·229	1160·625	2·824	670	·0112	..
5	10·180	162·330	162·722	7·855	1000·727	1163·449	2·377	447	·0137	..
6	12·216	170·123	170·577	6·848	995·249	1165·826	2·070	307	·0163	..
7	14·252	176·910	177·425	6·056	990·471	1167·896	1·830	240	·0189	..
8	16·288	182·910	183·481	5·460	986·245	1169·726	1·649	181	·0214	..

p	q	r	s	t	u	v	w	x	y	z
9	18·324	188·316	188·941	4·978	982·434	1171·375	1·502	147	·0239	..
10	20·360	193·240	193·919	4·577	978·958	1172·877	1·381	121	·0264	..
11	22·396	197·768	198·496	4·241	975·762	1174·258	1·279	102	·0289	..
12	24·432	201·960	202·737	3·972	972·800	1175·537	1·197	82	·0313	..
13	26·468	205·885	206·709	3·719	970·025	1176·734	1·121	76	·0337	..
14	28·504	209·560	210·428	3·511	967·427	1177·855	1·057	64	·0362	..
15	30·540	213·025	213·939	3·313	964·973	1178·912	·997	60	·0387	·304
16	32·576	216·296	217·252	3·157	962·657	1179·909	·950	47	·0413	1·304
17	34·612	219·410	220·409	3·010	960·450	1180·859	·905	45	·0437	2·304
18	36·648	222·378	223·419	2·866	958·345	1181·764	·864	41	·0462	3·304
19	38·684	225·203	226·285	2·754	956·343	1182·628	·826	38	·0487	4·304
20	40·720	227·917	229·039	2·637	954·415	1183·454	·792	34	·0511	5·304
21	42·756	230·515	231·676	2·542	952·570	1184·246	·763	29	·0536	6·304
22	44·792	233·017	234·218	2·454	950·791	1185·009	·735	28	·0561	7·304
23	46·828	235·432	236·672	2·357	949·072	·744	·709	26	·0585	8·304
24	48·864	237·752	239·029	2·285	947·424	1186·453	·686	23	·0610	9·304
25	50·900	240·000	241·314	2·212	945·825	1187·139	·664	22	·0634	10·304
26	52·936	242·175	243·526	2·145	944·277	·803	·643	21	·0658	11·304

27	54·972	244·284	245·671	2·077	942·775	1188·446	·623	20	·0683	12·304
28	57·008	246·326	247·748	2·021	941·321	1189·069	·605	18	·0707	13·304
29	59·044	248·310	249·769	1·969	939·905	·674	·589	16	·0731	14·304
30	61·080	250·245	251·738	1·910	938·925	1190·263	·5728	16	·0755	15·304
31	63·115	252·122	253·648	1·864	937·1878	·8358	·5580	148	·0779	16·304
32	65·152	253·952	255·512	1·817	935·8818	1191·3938	·5440	140	·0803	17·304
33	67·187	255·735	257·329	1·774	934·6088	·9378	·5310	130	·0827	18·304
34	69·224	257·476	259·103	1·732	933·3658	1192·4688	·5185	125	·0851	19·304
35	71·259	259·176	260·835	1·692	932·1523	·9873	·5065	120	·0875	20·304
36	73·296	260·835	262·527	1·655	930·9668	1193·4938	·4950	115	·0899	21·304
37	75·331	262·458	264·182	1·619	929·8068	·9888	·4840	110	·0922	22·304
38	77·378	264·045	265·801	1·585	928·6718	1194·4728	·4740	100	·0946	23·304
39	79·403	265·599	267·386	1·552	927·5608	·9468	·4640	100	·0970	24·304
40	81·440	267·120	268·938	1·522	926·4728	1195·4108	·4550	90	·0994	25·304
41	83·477	268·611	270·460	1·494	925·4058	·8658	·4460	90	·1017	26·304
42	85·512	270·073	271·954	1·463	924·3578	1196·3118	·4375	85	·1041	27·304
43	87·547	271·507	273·417	1·438	923·3323	·7493	·4295	80	·1064	28·304
44	89·584	272·915	274·855	1·411	922·3238	1197·1788	·4215	80	·1088	29·304

p	q	r	s	t	u	V	W	X	y	z
45	91·620	274·296	276·266	1·385	921·3343	1197·6003	·4139	76	·1111	30·304
46	93·656	275·652	277·651	1·365	920·3632	1198·0142	·4070	69	·1134	31·304
47	95·692	276·986	279·016	1·339	919·4052	·4212	·4000	70	·1158	32·304
48	97·729	278·297	280·355	1·317	918·4662	·8212	·3930	70	·1181	33·304
49	99·764	279·585	281·672	1·297	917·5422	1199·2142	·3864	66	·1204	34·304
50	101·801	280·854	282·969	1·274	916·6316	·6006	·3801	63	·1227	35·304
51	103·836	282·099	284·243	1·256	915·7377	·9807	·3740	61	·1251	36·304
52	105·873	283·326	285·499	1·237	914·8557	1200·3547	·3684	56	·1274	37·304
53	107·908	284·534	286·736	1·216	913·9871	·7231	·3629	55	·1297	38·304
54	109·945	285·724	287·952	1·201	913·1340	1201·0860	·3576	53	·1320	39·304
55	111·980	286·897	289·153	1·182	912·2906	·4436	·3525	51	·1343	40·304
56	114·027	288·052	290·335	1·168	911·4611	·7961	·3476	49	·1366	41·304
57	116·053	289·112	291·503	1·151	910·6407	1202·1437	·3428	48	·1388	42·304
58	118·089	290·316	292·654	1·136	909·8325	·4865	·3381	47	·1411	43·304
59	120·125	291·425	293·790	1·121	909·0346	·8246	·3336	45	·1434	44·304
60	122·160	292·520	294·911	1·105	908·2472	1203·1582	·3291	42	·1457	45·304
61	124·196	293·598	296·016	1·092	907·4713	·4873	·3249	42	·14793	46·304
62	126·232	294·663	297·108	1·077	906·7042	·8122	·3207	42	·15021	47·304

63	128·268	295·714	298·185	1·064	905·9477	1204·1329	·3166	41	·15248	48·304
64	130·304	296·752	299·249	1·051	905·2005	·4495	·3126	40	·15471	49·304
65	132·340	297·777	300·300	1·038	904·4621	·7621	·3086	40	·15697	50·304
66	134·376	298·789	301·338	1·026	903·7327	1205·0707	·3049	37	·15921	51·304
67	136·412	299·789	302·364	1·013	903·0116	·3756	·3013	36	·16147	52·304
68	138·448	300·776	303·377	1·003	902·2999	·6769	·2978	35	·16372	53·304
69	140·484	301·753	304·380	·990	901·5947	·9747	·2944	34	·16598	54·304
70	142·520	302·718	305·370	·980	900·8991	1206·2691	·2910	34	·16817	55·304
71	144·556	303·673	306·350	·970	900·2101	·5601	·2879	31	·17038	56·304
72	146·592	304·617	307·320	·959	899·5280	·8480	·2847	32	·17259	57·304
73	148·628	305·551	308·279	·949	898·8537	1207·1327	·2816	31	·17481	58·304
74	150·664	306·474	309·228	·938	898·1863	·4143	·2786	30	·17704	59·304
75	152·700	307·388	310·166	·926	897·5269	·6929	·2755	31	·17923	60·304
76	154·736	308·290	311·092	·919	896·8764	·9684	·2727	28	·18142	61·304
77	156·772	309·184	312·011	·909	896·2301	1208·2411	·2699	28	·18360	62·304
78	158·808	310·069	312·920	·901	895·5910	·5110	·2671	28	·18579	63·304
79	160·844	310·945	313·821	·891	894·9571	·7781	·2643	28	·18797	64·304
80	162·880	311·812	314·712	·883	894·3304	1209·0424	·2618	25	·19015	65·304

p	q	r	s	t	u	v	w	x	y	z
81	164·916	312·670	315·595	·873	893·7092	1209·3042	·2592	26	·19232	66·304
82	166·952	313·520	316·468	·865	893·0954	·5634	·2567	25	·19454	67·304
83	168·988	314·361	317·333	·857	892·4871	·8201	·2542	25	·19674	68·304
84	171·024	315·195	318·190	·850	891·8843	1210·0743	·2519	23	·19887	69·304
85	173·060	316·021	319·040	·842	891·2862	·3262	·2496	23	·20105	70·304
86	175·096	316·839	319·882	·835	890·6938	·5758	·2473	23	·20321	71·304
87	177·132	317·650	320·717	·826	890·1061	·8231	·2450	23	·20535	72·304
88	179·168	318·453	321·543	·819	889·5251	1211·0681	·2429	21	·20753	73·304
89	181·204	319·249	322·362	·814	888·9490	·3110	·2408	21	·20970	74·304
90	185·240	320·039	323·176	·805	888·3758	·5518	·2386	21	·21183	75·304
91	185·276	320·821	323·981	·800	887·8094	·7904	·2366	20	·21393	76·304
92	187·312	321·597	324·781	·791	887·2460	1212·0270	·2346	20	·21608	77·304
93	189·348	322·366	325·572	·786	886·6896	·2616	·2326	20	·21829	78·304
94	191·384	323·128	326·358	·778	886·1362	·4942	·2305	21	·22045	79·304
95	193·422	323·884	327·136	·773	885·5887	·7247	·2287	18	·22247	80·304
96	195·456	324·634	327·909	·766	885·0444	·9534	·2268	19	·22455	81·304
97	197·492	325·378	328·675	·758	884·5052	1213·1802	·2249	19	·22667	82·304
98	199·528	326·114	329·433	·753	883·9721	·4051	·2230	19	·22883	83·304
99	201·564	326·845	330·186	·749	883·4421	·6281	·2213	17	·23095	84·304

100	203·600	327·571	330·935	·743	882·9144	·8494	·2195	18	·23302	85·304
101	205·636	328·291	331·678	·736	882·3909	1214·0689	·2178	17	·23510	86·304
102	207·672	329·005	332·414	·731	881·8727	·2867	·2160	18	·23717	87·304
103	209·708	329·714	333·145	·724	881·3577	·5027	·2144	16	·23925	88·304
104	211·744	330·416	333·869	·718	880·8481	·7171	·2128	16	·24132	89·304
105	213·780	331·113	334·587	·714	880·3429	·9299	·2111	17	·24340	90·304
106	215·816	331·805	335·301	·708	879·8400	1215·1410	·2096	15	·24547	91·304
107	217·852	332·492	336·009	·705	879·3416	·3506	·2081	15	·24754	92·304
108	219·888	333·174	336·714	·697	878·8447	·5587	·2065	16	·24961	93·304
109	221·924	333·851	337·411	·694	878·3542	·7652	·2051	14	·25168	94·304
110	223·960	334·523	338·105	·690	877·8653	·9703	·2036	15	·25376	95·304
111	225·996	335·191	338·795	·684	877·3789	1216·1739	·2021	15	·25582	96·304
112	228·032	335·854	339·479	·678	876·8970	·3760	·2008	13	·25788	97·304
113	230·068	336·511	340·157	·675	876·4198	·5768	·1994	14	·25994	98·304
114	232·104	337·165	340·830	·670	875·9442	·7762	·1979	15	·26199	99·304
115	234·140	337·814	341·502	·667	875·4721	·9741	·1967	12	·26405	100·304
116	236·176	338·459	342·169	·662	875·0018	1217·1708	·1954	13	·26611	101·304
117	238·212	339·100	342·831	·657	874·5352	·3662	·1940	14	·26816	102·304

p	q	r	s	t	u	v	w	x	y	z
118	240·248	339·736	343·488	·653	874·0722	1217·5602	·1928	12	·27020	103·304
119	242·284	340·368	344·141	·648	873·6120	·7530	·1915	13	·27224	104·304
120	244·320	340·995	344·789	·643	873·1555	·9445	·1902	13	·27428	105·304
121	246·356	341·618	345·432	·641	872·7027	1218·1347	·1891	11	·27628	106·304
122	248·392	342·238	346·073	·636	872·2508	·3238	·1879	12	·27828	107·304
123	250·428	342·854	346·709	·634	871·8027	·5117	·1866	13	·28027	108·304
124	252·464	343·466	347·343	·629	871·3553	·6983	·1855	11	·28227	109·304
125	254·500	344·074	347·972	·624	870·9118	·8838	·1843	12	·28426	110·304
126	256·536	344·678	348·596	·621	870·4721	1219·0681	·1831	12	·28625	111·304
127	258·572	345·279	349·217	·618	870·0342	·2512	·1821	10	·28824	112·304
128	260·608	345·876	349·835	·613	869·5983	·4333	·1810	11	·29023	113·304
129	262·644	346·459	350·448	·611	869·1663	·6143	·1798	12	·29222	114·304
130	264·680	347·059	351·059	·606	868·7351	·7941	·1788	10	·29420	115·304
131	266·716	347·644	351·665	·602	868·3079	·9729	·1777	11	·29618	116·304
132	268·752	348·227	352·267	·600	867·8836	1220·1506	·1765	12	·29816	117·304
133	270·788	348·806	352·867	·596	867·4601	·3271	·1756	9	·30013	118·304
134	272·824	349·382	353·463	·592	867·0397	·5027	·1746	10	·30209	119·304
135	274·860	349·954	354·055	·589	866·6223	·6773	·1735	11	·30405	120·304

136	276·896	350·523	354·644	·586	866·2068	·8508	·1726	9	·30601	121·304
137	278·932	351·089	355·230	·583	866·7934	1221·0234	·1716	10	·30796	122·304
138	280·968	351·752	355·813	·579	865·3820	·1950	·1705	11	·30990	123·304
139	283·004	352·211	356·392	·577	864·9735	·3655	·1696	9	·31186	124·304
140	285·040	352·767	356·969	·572	864·5661	·5351	·1686	10	·31386	125·304
141	287·076	353·319	357·541	·569	864·1627	·7037	·1676	10	·31587	126·304
142	289·112	353·869	358·110	·567	863·7613	·8713	·1668	8	·31788	127·304
143	291·148	354·416	358·677	·563	863·3611	1222·0381	·1659	9	·31990	128·304
144	293·184	354·960	359·240	·561	862·9640	·2040	·1649	10	·32190	129·304
145	295·220	355·501	359·801	·558	862·5679	·3689	·1641	8	·32391	130·304
146	297·256	356·039	360·359	·554	862·1740	·5330	·1632	9	·32592	131·304
147	299·292	356·574	360·913	·552	861·7832	·6962	·1622	10	·32794	132·304
148	301·328	357·106	361·465	·548	861·3934	·8584	·1614	8	·32995	133·304
149	303·364	357·635	362·013	·546	861·0068	1223·0198	·1605	9	·33196	134·304
150	305·400	358·161	362·559	·541	860·6213	·1803	·1596	9	·33400	135·304
151	307·436	358·683	363·100	·540	860·2399	·3399	·1589	7	·33580	136·304
152	309·472	358·203	363·640	·537	859·8588	·4988	·1581	8	·33761	137·304
153	311·508	359·721	364·177	·534	859·4799	·6569	·1572	9	·33942	138·304

p	q	r	s	t	u	v	w	x	y	z
154	313·544	360·236	364·711	·532	859·1031	1223·8141	·1565	7	·34123	139·304
155	315·580	360·749	365·243	·530	858·7276	·9706	·1557	8	·34304	140·304
156	317·616	361·260	365·773	·527	858·3533	1224·1263	·1548	9	·34485	141·304
157	319·652	361·768	366·300	·524	857·9811	·2811	·1541	7	·34666	142·304
158	321·688	362·273	366·824	·523	857·6112	·4352	·1533	8	·34847	143·304
159	323·724	362·776	367·347	·520	857·2415	·5885	·1525	8	·35028	144·304
160	325·760	363·277	367·867	·516	856·8740	·7410	·1519	6	·35209	145·304
161	327·796	363·774	368·383	·515	856·5099	·8929	·1512	7	·35397	146·304
162	329·832	364·270	368·898	·512	856·1461	1225·0441	·1505	7	·35585	147·304
163	331·868	364·764	369·410	·510	855·7846	·1946	·1497	8	·35773	148·304
164	333·904	365·255	369·920	·508	855·4243	·3443	·1491	6	·35961	149·304
165	335·940	365·744	370·428	·506	855·0654	·4934	·1483	8	·36149	150·304
166	337·776	366·232	370·934	·504	854·7077	·6417	·1477	6	·36337	151·304
167	340·012	366·717	371·438	·501	854·3514	·7894	·1470	7	·36525	152·304
168	342·048	367·199	371·939	·498	853·9974	·9364	·1462	8	·36714	153·304
169	344·084	367·680	372·437	·497	853·6456	1226·0826	·1456	6	·36903	154·304
170	346·120	368·158	372·934	·493	853·2942	·2282	·1449	7	·37092	155·304
171	348·156	368·632	373·427	·491	852·9461	·3731	·1444	5	·37272	156·304

172	350·192	369·105	373·918	·490	852·5995	·5175	·1438	6	·37452	157·304
173	352·228	369·576	374·408	·487	852·2533	·6613	·1431	7	·37632	158·304
174	354·264	370·045	374·895	·485	851·9094	·8044	·1426	5	·37812	159·304
175	356·300	370·512	375·380	·485	851·5670	·9470	·1419	7	·37992	160·304
176	358·336	370·978	375·865	·482	851·2239	1227·0889	·1414	5	·38172	161·304
177	360·372	371·442	376·347	·480	850·8833	·2303	·1408	6	·38353	162·304
178	362·408	371·904	376·827	·478	850·5441	·3711	·1401	7	·38534	163·304
179	364·444	372·364	377·305	·476	850·2062	·5112	·1396	5	·38715	164·304
180	366·480	372·822	377·781	·474	849·8698	·6508	·1389	7	·38895	165·304
181	368·516	373·275	378·255	·472	849·5347	·7897	·1384	5	·39077	166·304
182	370·552	373·731	378·727	·470	849·2011	·9281	·1378	6	·39259	167·304
183	372·588	374·183	379·197	·468	848·8689	1228·0659	·1371	7	·39441	168·304
184	374·624	374·633	379·665	·466	848·5380	·2030	·1366	5	·39624	169·304
185	376·660	375·081	380·131	·464	848·2086	·3396	·1359	7	·39807	170·304
186	378·696	375·527	380·595	·461	847·8805	·4755	·1354	5	·39990	171·304
187	380·732	375·971	381·056	·460	847·5549	·6109	·1348	6	·40173	172·304
188	382·768	376·413	381·516	·458	847·2297	·7457	·1341	7	·40356	173·304
189	384·804	376·853	381·974	·455	846·9058	·8798	·1336	5	·40539	174·304
190	386·840	377·291	382·429	·454	846·5844	1229·0134	·1329	7	·40722	175·304

	q	r	s	t	u	v	w	x	y	z
191	388·876	377·727	382·883	·452	846·2633	1229·1463	·1324	5	·40899	176·304
192	390·912	378·161	383·335	·450	845·9437	·2787	·1319	5	·41076	177·304
193	392·948	378·593	383·785	·448	845·6256	·4106	·1313	6	·41253	178·304
194	394·984	379·023	384·233	·446	845·3089	·5419	·1309	4	·41430	179·304
195	397·020	379·452	384·679	·444	844·9938	·6728	·1303	6	·41607	180·304
196	399·056	379·979	385·123	·444	844·6801	·8031	·1299	4	·41784	181·304
197	401·092	380·305	385·567	·441	844·3660	·9330	·1293	6	·41962	182·304
198	403·128	380·729	386·008	·441	844·0543	1230·0623	·1289	4	·42140	183·304
199	405·164	381·152	386·449	·438	843·7422	·1912	·1284	5	·42318	184·304
200	407·200	381·573	386·887	·437	843·4326	·3196	·1278	6	·42496	185·304
201	409·236	381·992	387·324	·436	843·1234	·4474	·1274	4	·42667	186·304
202	411·272	382·410	387·760	·434	842·8148	·5748	·1268	6	·42838	187·304
203	413·308	382·827	388·194	·433	842·5076	·7016	·1264	4	·43009	188·304
204	415·344	383·242	388·627	·430	842·2010	·8280	·1259	5	·43180	189·304
205	417·380	383·655	389·057	·428	841·8969	·9539	·1253	6	·43351	190·304
206	419·416	384·066	389·485	·427	841·5942	1231·0792	·1249	4	·43523	191·304
207	421·452	384·475	389·912	·425	841·2921	·2041	·1243	6	·43695	192·304
208	423·488	384·883	390·337	·422	840·9914	·3284		4	·43866	193·304

209	425·524	385·288	390·759		840·6933	·4523		5	·44039	194·304
				·420			·1239			
210	427·560	385·671	391·179		840·3967	·5757		..	·44211	195·304
				4·143			·1234			
220	447·920	389·631	395·322		837·483	1232·805		..	·4591	205·304
				4·001			1·2293			
230	468·280	393·471	399·323		834·650	1233·973		..	·4759	215·304
				3·862			1·168			
240	488·640	397·192	403·185		831·899	1235·084		..	·4926	225·304
				3·732			1·111			
250	509·000	400·793	406·917		829·230	1236·147		..	·5090	235·304
				3·660			1·063			
260	529·360	404·275	410·577		826·641	1237·218		..	·5253	245·304
							1·071			

TABLE XI.

The Properties of Saturated Steam, at Pressures from One Atmosphere to Eighteen Atmospheres.

Elastic Force. In Atmospheres.	Elastic Force. In pounds on the square inch.	Elastic Force. In inches of mercury at 32°.	Temperature in degrees Fahrenheit of the steam, and of the water from which it is evaporated.	B.T.U. per lb. from zero Fahrenheit: Number contained in the water.	Difference.	Number required for evaporation, known as latent heat, or heat of vaporization.	Total number contained in the steam.	Difference.	2nd Difference.	Weight of one cubic foot, in decimals of a pound.
·25	3·674	7·480	149·649	149·943	29·598	1009·6372	1159·5802	8·9548		·01032
·5	7·348	14·961	179·008	179·541	19·176	988·9940	1168·5350	5·7910	3·1638	·01976
·75	11·022	22·441	197·987	198·717	14·183	975·6090	1174·3260	4·2740	1·5170	·02912
1·	14·696	29·922	212·000	212·900	21·425	965·7000	1178·6000	6·4425		·03797
1·5	22·044	44·883	233·123	234·325	16·213	950·7175	1185·0425	4·8632	1·5793	·05616
2·	29·392	59·844	249·068	250·538	13·212	939·3677	1189·9057	3·9553	·9079	·07415
2·5	36·740	74·805	262·036	263·750	11·228	930·1110	1193·8610	3·3553	·6000	·09166
3·	44·088	89·766	273·037	274·978	9·811	922·2383	1197·2163	2·9271	·4282	·10894
3·5	51·436	104·727	282·634	284·789	8·753	915·3544	1200·1434	2·6080	·3191	·12610

4·	58·784	119·688	291·185	293·542	7·931	909·2094	1202·7514	2·3595	·2485	·14284
4·5	66·132	134·649	298·921	301·473	7·260	903·6379	1205·1109	2·1585	·2010	·15949
5·	73·480	149·610	305·994	308·733	6·708	898·5364	1207·2694	1·9898	·1687	·17584
5·5	80·828	164·571	312·522	315·441	6·244	893·8182	1209·2592	1·8517	·1381	·19195
6·	88·176	179·532	318·593	321·685	5·854	889·4259	1211·1109	1·7335	·1182	·20780
6·5	95·524	194·493	324·277	327·539	5·509	885·3054	1212·8444	1·6306	·1029	·22350
7·	102·872	209·454	329·623	333·048	5·206	881·4270	1214·4750	1·5393	·0913	·23899
7·5	110·220	224·415	334·670	338·254	4·947	877·7603	1216·0143	1·4610	·0783	·25417
8·	117·568	239·376	339·461	343·201	4·715	874·2743	1217·4753	1·3914	·0696	·26906
8·5	124·916	254·337	344·023	347·916	4·509	871·9507	1218·8667	1·3292	·0622	·28387
9·	132·264	269·298	348·380	352·425	4·318	867·7709	1220·1959	1·2722	·0570	·29852
9·5	139·612	284·259	352·551	356·743	4·145	864·7251	1221·4681	1·2205	·0517	·31290
10·	146·960	299·220	356·553	360·888	7·830	861·8006	1222·6886	2·3019		·32718
11·	161·656	329·142	364·099	368·718	7·314	856·2725	1224·9905	2·1475	·1544	·35494
12·	176·352	359·064	371·141	376·032	6·870	851·1060	1227·1380	2·0151	·1324	·38227
13·	191·048	388·986	377·748	382·902	6·468	846·2511	1229·1531	1·8947	·1204	·40880
14·	205·744	418·908	383·960	389·370	6·136	841·6778	1231·0478	1·7934	·1013	·43441
15·	220·440	448·830	389·840	395·506	5·834	837·3352	1232·8412	1·7046	·0888	·45934
16·	235·136	478·752	395·429	401·340	5·572	833·2058	1234·5458	1·6245	·0801	·48390
17·	249·832	508·674	400·755	406·912	5·334	829·2583	1236·1703	1·5515	·0730	·50875
18·	264·528	538·596	405·842	412·246		825·4758	1237·7218			·53446

CHAPTER XVIII.

THE ACTION AND TREATMENT OF STEAM IN A STEAM-ENGINE.

Production and utilization of steam. — The useful power of steam depends upon the increase in volume which is assumed by water under the influence of an application of heat, whereby the volume of steam emitted by the boiler is greater than that of the liquid water supplied. The steam is utilized in one or more cylinders of a steam-engine, in each of which it performs work by virtue of its pressure upon a moving piston. In this operation the steam necessarily loses heat in proportion to the amount of work performed.

Properties of steam.—The density, temperature, and other properties of steam at any ordinary pressure may be obtained from Tables IX., X., and XI., which by permission of Messrs. Elliott are extracted from Porter's work on the Richards Indicator. These figures refer to steam which is in free contact with water, or is only just evaporated, in which condition it is termed "saturated steam." This is of one invariable temperature, corresponding to any particular pressure. On the application of heat to steam which is not in contact with water its temperature is increased, and it becomes "superheated." If during this operation the steam is

allowed to expand freely, its pressure may remain constant; but if confined to a constant volume, its pressure must rise.

Condensation of saturated steam by loss of heat.—When saturated steam, at a constant volume, is exposed to any kind of cooling influence, a portion is condensed. This may arise from radiation, in which case the condensation will be distributed throughout the mass; or from abstraction of heat by conduction, in which case the condensation will chiefly take place upon the walls of the vessel containing the steam, or upon such portion of the walls as assume a lower temperature than that of the steam. In the latter case, the condensed water will collect upon the walls of the vessel as a mist, such as is often seen on a window-pane, and which really consists of a series of minute drops. If the cooling influence ceases, and heat is applied, the mist or film is re-evaporated, and this may be repeated an indefinite number of times. So long as any portion of the original steam remains in a condensed condition, the pressure in the vessel is less than it was before the condensation, or after complete re-evaporation. If the condensation is continued, the increased amount of water upon the surface causes the minute drops to coalesce, when they may be observed by the adoption of suitable means. A further continuance of the condensation increases the size of the drops so much that they drain to the lowest part of the surface, and may fail to be re-evaporated in the interval before the next period of condensation. When this is the case the water accumulates, and requires to be removed by drainage.

Condensation by conversion of heat into work.—When cooling arises from the performance of mechanical work of saturated steam, a partial condensation of the steam

usually occurs. The moisture thus arising may remain diffused in the mass, but will be rapidly precipitated upon any surface in contact with it, and of a lower temperature. Moisture produced in this way is in addition to that produced by physical cooling, and can only be re-evaporated by the application from an external source of additional heat corresponding to the amount of work done, or by the expenditure of mechanical power equal in amount to that originally produced.

Apportionment of heat expended in raising steam.— Heat is imparted to water with the object of the conversion into mechanical work of the largest possible proportion of the heat expended. A portion of the heat is expended as sensible heat, in raising the temperature of the water, but the remaining and very much larger portion is expended as latent heat of evaporation in the direct production of steam, or in changing the water from a liquid to a gaseous condition.

Apportionment of heat in the application of steam.— In the subsequent utilization of the steam, a very large proportion of the sensible heat is transformed into work. But the amount of work done is usually found to be several times greater than the sensible heat alone can yield. The balance is furnished by the latent heat liberated in the condensation of a portion of the steam in the cylinder, with or without the addition of heat from external sources. Assuming that 15 pounds of saturated steam, at an absolute pressure of 165 pounds per square inch, are required to produce one horse-power, by expansion to an absolute pressure of 3 pounds per square inch, and neglecting all minor losses, it will be found that about 40 per cent. of the power will be produced by the sensible heat corresponding to the limits of temperature due to the given pressures. Also that the remaining 60 per cent. of the power is due to

the condensation of 10 per cent. of the total quantity of steam supplied. This condensation may be effected in the mass of steam in the cylinder, or it may take place in a steam-jacket, but it is quite impossible to prevent it by any means which are practically available in connection with the use of saturated steam. Probably, even in the best condensing engines, no more than 15 per cent. of the latent heat of the steam is utilized for the production of power. The remainder of the latent heat, amounting to 78 to 80 per cent. of the total heat supplied to the engine, is therefore left in the steam, after it has performed all the work of which under the conditions it is capable. This heat is incorporated in a small amount of steam, but in the act of condensation it becomes incorporated as sensible heat in a large amount of water, the whole of which is passed away by the air-pump. The number of thermal units lost in this way per indicated horse-power per minute is Farey and Donkin's co-efficient, and is an important element in the comparison between different engines, or between the same engine when working under different conditions. According to the type and condition of the engine, this co-efficient varies from 200 to 500 in tested engines, a low co-efficient indicating high efficiency. The amount of heat actually or usefully consumed in the production of one horse-power is 42·75 units per minute, so that of the whole amount of heat supplied to the engine of the boiler, the percentage lost =

$$\frac{100 \times \text{co-efficient}}{42{\cdot}75 + \text{co-efficient.}}$$

The percentage utilized =

$$\frac{100 \times 42{\cdot}75}{42{\cdot}75 + \text{co-efficient.}}$$

The large proportion of heat lost is shown again in the coal consumption, of which only 0·2 pound of good

average quality per indicated horse-power per hour are due to the actual work performed. Some of the balance of heat passes away in the furnace gases, and is lost in other ways in connection with the boiler, but by far the greater part passes away to the condenser, and is rejected in the air-pump discharge. When a fresh and constant supply of condensing water is available, the rejected heat is at once completely disposed of. When, owing to a scanty supply, the water must be repeatedly used, the imparted heat must be removed from it, so as to approximately restore the original temperature. A cooling-pond is generally employed for this purpose, but other appliances may be entirely or partially substituted for it, as described in another chapter. An air-condenser, with or without assistance from surface evaporation, may also be adopted, in which case the rejected heat is directly dispersed in the atmosphere.

Expansion of steam of high pressure.—Steam may be utilized in an engine either by means of its direct pressure, supported by a continuous supply from the boiler, or it may be supplied to the engine in an intermittent manner, and caused to act by its force of expansion during the intervals in which the supply is cut off. Steam of all pressures may be applied expansively, and thereby caused to perform some work, but the work to be thus derived from steam of low pressure is relatively so small as to render its use unprofitable. Steam of high pressure is, however, able to perform a very large amount of work during expansion. One pound of steam of a pressure of 185 pounds per square inch above the atmosphere absorbs in its production about 4 per cent. more heat than does steam of 5 pounds pressure. The former will perform about 22 per cent. more work when used non-expansively; but

it will theoretically perform 140 per cent. more work when expanded in a condensing engine to an absolute pressure of 5 pounds, or about 10 pounds below the atmosphere; and when practically utilized in a multiple expansion engine the increase is greater still. The advantage to be secured by the use of steam of 85 pounds pressure is about two-thirds of the above. One pound of steam will perform an approximately equal amount of work during expansion in a given proportion, whatever is the initial pressure, and provided that the back pressure bears in each case the same proportion to the initial pressure. Thus, steam of 100 pounds expanded four times, and working to a back pressure of 25 pounds, will perform approximately the same amount of work as the same weight of steam of 20 pounds pressure, expanded four times and working against a back pressure of 5 pounds. Though the full advantage to be derived from the use of high-pressure steam can only be secured by the adoption of expansive working, an indirect advantage can sometimes be secured by reason of the use of an engine of smaller size, and usually smaller cost than would be necessary with steam of low pressure.

Expansion of steam, without and with the performance of mechanical work.—All perfect gases may be expanded in such a manner as to avoid performing any work in the operation. When this condition is observed—as for instance when gas is allowed to flow into an exhausted receiver—the gas possesses no inherent tendency to vary in temperature, and its absolute pressure at any moment varies inversely in accordance with the volume occupied. If successive pressures are laid off upon an indicator diagram, corresponding to the respective volumes, a hyperbolic curve will be obtained, which is also known as a curve of isothermal

expansion, or expansion at a uniform temperature. When steam is caused to expand to perform work, heat disappears, and the temperature falls. According to the expansion of a perfect gas, steam in expansion might be expected to fall in pressure more rapidly than in accordance with the inverse proportion of volume. In the expansion of saturated steam, however, when performing mechanical work, the pressure varies so nearly in accordance with the hyperbolic curve, that this curve is a very convenient standard for use in the analysis of indicator diagrams. If the expansion curve upon the diagram departs largely from the hyperbolic curve, leakage of the piston or valves probably exists, or an abnormal amount of condensation and re-evaporation prevails.

Condensation and re-evaporation of steam in cylinder due to variation in cylinder temperature.—In all working cylinders the steam-pressure must of necessity be considerably higher during the time of admission than during exhaust, and a definite difference in the temperature of the steam follows from this. The difference in steam temperature is approximately imparted to the surfaces of the cylinder, which are in contact with it, including those of the cylinder bore or shell, covers, piston, piston-rod, ports, and valves. These receive heat during the time of admission of steam, and are cooled by the abstraction of heat during exhaust. The addition is effected by the condensation upon the surfaces of such particles of fresh steam as may be in contact with them, the latent heat of which is liberated; and, conversely, the abstraction is effected by the re-evaporation of the water previously condensed. The steam of reduced temperature exercises a cooling action upon the surfaces after the water is exhausted, but if the steam is nearly dry this process is slow. In an

ordinary cylinder, supplied with saturated steam, condensation always occurs during the period of admission. Efficient protection afforded by non-conducting coverings will cause a reduction in cylinder condensation, but will not prevent it. After cut-off, the steam being allowed to expand, the pressure falls. The temperature of the cylinder walls and of the condensed water attached to them then exceeds the temperature of saturation corresponding to the pressure, and the condensed water is re-evaporated, absorbing heat in the process from the cylinder walls, by which the temperature of the skin next to the steam falls very considerably. Re-evaporation of the condensed water commences in a simple engine immediately after cut-off, and continues or increases during the period of expansion, but is not completed before the end of the stroke. In all ordinary cases a considerable proportion of the condensed water remains to be re-evaporated during the period of exhaust. From the moment of condensation to that of re-evaporation, the useful power or active force of the steam so condensed is in suspense. If re-evaporation is effected before the end of the stroke, the power of such steam is restored. But steam which is condensed during admission, and is only re-evaporated during exhaust, performs no work, and is practically wasted. Condensed water, once deposited upon a surface, probably facilitates the action, but it appears to be initiated purely by a temperature in the walls, inferior to that of the steam at the time. In extreme cases the amount re-evaporated falls slightly below that condensed, in which case accumulation of water takes place, and must be continuously withdrawn. The presence of such condensed water upon the plane horizontal surfaces of the piston and lower cover of a vertical engine is especially pernicious, on account of difficulty in drainage.

Waste of heat by cylinder condensation. — When internal condensation takes place upon the surfaces, heat is withdrawn from steam of a high temperature which has had no opportunity to perform mechanical work. A very important proportion of this heat is only restored to the steam during the period of exhaust, and fulfils no practical purpose except raising the temperature of the air-pump discharge; hence the value of the results of measurement of the heat rejected in the air-pump discharge under different conditions.

Conditions affecting cylinder condensation.—Cylinder condensation and re-evaporation depend upon the alternate transfer of heat into and out of the cylinder surfaces, and may be checked by the adoption of any measures which impede such transfer.

The loss from cylinder condensation varies according to the difference of temperature between the cylinder and the steam. The temperature of the cylinder at the time of admission depends upon the temperature of the steam during the period of exhaust. The amount of condensation therefore depends ultimately upon the difference between the highest and the lowest temperatures of steam in the cylinder, unless modified by the supply of heat from without.

Range of temperature in each cylinder reduced by compound working.—Steam may be expended expansively in one cylinder or in several cylinders in succession. With the same degree of theoretical expansion in each case, the amount of work performed is the same. But by the use of several cylinders the range of temperature is divided with some approach to equality in each, so that condensation is reduced. In a compound, triple, or quadruple expansion engine, the capacity of the last cylinder in the series is precisely equal to that of the single cylinder which would be

required for simple expansion of the same quantity of steam expanded to the same degree. The capacities of the earlier cylinders are much less, so that much less surfaces are exposed to the hot steam, and the condensation in each is much less than proportionate to the work done.

The total number of expansions in an engine may be obtained by dividing the volume of the steam in the last cylinder at the end of the stroke by that in the first cylinder up to the point of cut-off. In each case the calculated volume should accurately include clearance space and allow for piston-rod, and when one stage of expansion is divided over two cylinders, the combined volume must be taken. The total number of expansions may also be obtained by multiplying the number of expansions in the first cylinder by the successive ratios of cylinder volume. In the case of a compound engine, in the high-pressure cylinder of which steam is cut off at such a point that

$$\frac{\text{total volume, including clearance}}{\text{volume at cut-off, including clearance}} = 3,$$

and whose cylinder ratio, making allowance for piston-rods and clearances, is 1 : 3·0, the total number of expansions is 9·0. In a triple expansion engine, with 2·5 expansions in the high-pressure cylinder, and whose successive cylinder ratios are each 1 : 2·5, the total number of expansions becomes 15·62. In each case the comparatively low range of temperature in each cylinder, and the moderate degree of expansion in each, combine to make the total loss by condensation in the two or three cylinders much less than it would be in the one cylinder of a simple engine. The advances in practice from compounds to triples, and thence to quadruples, have been much assisted by the fact that the pressure

of steam advances much more rapidly than the temperature. Thus the difference in temperature between steam at atmospheric pressure and at 50 pounds above is 85° F., while the difference between that of 150 pounds and 200 pounds is only 22° F. Therefore with ordinary pressures in each case the temperature range in quadruple engines is less than in compounds, while triple expansion engines are intermediate. The mean range of temperature in each of four cylinders, using steam of 230 pounds per square inch above the atmosphere, is only about the same as that in a simple condensing engine using steam at atmospheric pressure.

Cylinder condensation as affected by speed. — The cylinder condensation per stroke is less when the engine is run at a high speed of revolution. This is because the alternations of temperature cannot penetrate so far into the substance of the cylinder as is possible at a slower speed, and therefore less heat is consumed in the operation. Subject to minor corrections dealt with in another chapter, the amount of work done by an engine varies in accordance with the speed. Therefore, as regards the work done, the loss by condensation is reduced by an increase of speed. Probably this is almost exactly in proportion to $\frac{1}{\sqrt{\text{speed of revolution}}}$.

Cylinder condensation as affected by amount of cylinder surface exposed.—The loss arising from cylinder condensation may be limited by any means whereby the receptive power of the surfaces for heat may be diminished. This may be effected by providing a minimum of surface as compared with volume, so that the mean depth of steam $\left(=\frac{\text{volume}}{\text{surface}}\right)$ at the moment of greatest condensation may be as large as possible. Generally, this condition is most completely fulfilled by engines

whose cylinders are of small diameter as compared with the length of stroke, or in which the length of steam at cut-off is nearly equal to the diameter of cylinder, subject to modification on account of surface exposed by the ports and valves.

Cylinder condensation as affected by character of cylinder surfaces.—The same object is also secured by the adoption of any modification in the surfaces exposed to steam, whereby their powers of conduction and absorption or radiation are reduced. The application of this principle to frictional surfaces is very largely limited by the necessity for a hard tough surface to resist wear. Mr. Donkin has lately shown that a continuous film of oil applied to such surfaces is physically attended by excellent results, but at an extravagant cost. Whether for this purpose any means may be discovered whereby oil may be thus employed, and continuously utilized in a manner corresponding to the pump lubrication of a heavy bearing, it is impossible to say. As to the non-frictional surfaces there is less difficulty. Lead, wood, and other attached coverings have been applied to pistons and cylinder-covers by Smeaton and others. In 1890 Professor Thurston proposed a treatment of such surfaces by corrosive disintegration by the use of dilute acid, after which the pores were filled with a drying oil and completely covered. By this means a saving of 40 per cent. in the loss by internal cylinder condensation was claimed, and one of 50 per cent. anticipated, equal to a saving of 10 to 20 per cent. in the total amount of steam consumed by the engine. The usual coating of grease, which is found upon such surfaces after a period of use, must also act more or less in the same way.

Cylinder condensation as affected by temperature of material behind surfaces, and the use of a steam-jacket.—

The receptive power of the surfaces for heat is largely dependent upon the temperature prevailing in the mass of the substance immediately in the rear of the surface. When the inner surface of a cylinder shell is exposed to steam inside, and to the atmosphere (with or without the interposition of a cleading of non-conducting material) on the outside, it follows that the temperature of the inside will be higher than that of the outside. The opposite condition prevails if the shell of the cylinder is surrounded by steam of a temperature higher than that of the working steam in the cylinder. When a jacket is provided for the application of hot steam in this way, it acts by the direct supply of heat to the steam in the cylinder, by means of conduction through the shell, and also by interposing a resistance to the escape of heat through the shell. The alternate heating and cooling of the inner skin of the shell is also impeded by reason of the fact that this skin is surrounded by metal of a higher temperature than itself, while, in the absence of a jacket, the opposite condition must prevail. The surface cannot become so thoroughly cooled, therefore, its power for condensation is impaired, and the jacket caused to act as a passive rather than as an active agent. Probably the last-named service is the most important which a steam-jacket renders in connection with a simple engine, or the first cylinder of a series, in each of which an unlimited supply of steam from the boiler to the cylinder is available. But in the later cylinders of a series in which the supply of steam is limited to that contained in the intermediate pipes or receivers, the condensation upon the cylinder surfaces, during the time of admission, is also limited. But though such condensation is reduced in amount, it is continued into the period of expansion after cut-off. It is therefore more susceptible to restric-

tion by means of the direct supply of heat, from jackets, to the steam within the cylinder. In each case the benefit derived from the use of a jacket is not measured, even approximately, by the amount of heat actually supplied by the jacket to the steam in the cylinder. In each case the chief duty of the jacket is to modify or regulate the distribution of the heat of the working steam in the cylinder, so that it is always so disposed that the largest possible proportion is transformed into mechanical work, and the least possible proportion allowed to evade its duty by disappearance in the substance of the cylinder only to reappear and escape by the exhaust. Heat which is thus allowed to escape from the last cylinder of a series is totally lost, but heat which thus escapes from the preceding cylinder or cylinders may possibly become usefully treated in the next cylinder. The last cylinder of the series is therefore the one in which the greatest amount of advantage can be derived from the use of a jacket. The use of a steam-jacket upon the low-pressure cylinder is sometimes condemned, under an impression that the heat thus supplied is largely and wastefully devoted to heating the exhaust steam. This would obviously be objectionable, as it would increase the amount of heat rejected in the exhaust, which it is the object of all such measures to diminish. But the common experience is that the use of efficient jackets is followed by a reduction in the heat rejected by the condenser.

Cylinder condensation during admission of steam from unlimited supply.—The condensation of steam which occurs during the period of admission would cause a reduction in the pressure which prevails inside the cylinder, if the quantity of steam were strictly limited. But the pressure is maintained by reason of communi-

cation with the boiler, the supply from which is practically unlimited. The power of the surfaces to effect condensation, by reason of their reduced temperature, is therefore probably spent before the point of cut-off is reached.

Supply of steam limited by receiver.—If the steam is passed into a second or third cylinder for use at lower pressure, in a compound or triple expansion engine, the supply is not unlimited. The condensation which first takes place causes a fall in temperature and pressure which retards subsequent condensation. This is protracted into the period of expansion, and probably to near the end of the stroke. This delay in condensation causes a reduction in its amount. The increased time during which condensation is effected in such a case, as compared with the former case, leads to a greater demand for steam in the jacket. More steam is required and more benefit derived from its use in the jackets of such cylinders than in those of first cylinders or simple cylinders with equal cut-off. The large number of variable elements affecting this question, however, quite prevent the application of any accurate calculations.

Condensation due to performance of work.—In addition to the condensation which occurs as a direct consequence of changes in the temperature of the cylinder walls, which condensation is confined to the cylinder surfaces, and is of uncertain amount, condensation also arises from the performance of mechanical work, as already explained. This is uniformly distributed throughout the mass of expanding steam. The condensed water thus arising facilitates the transfer of heat, rendering the steam more prone to condensation upon the surfaces, and therefore increases the duty required from the steam-jacket. Probably in no case, however, can the whole of the water so condensed, or even the most of

it, become separated from the main body of steam and deposited upon the surfaces.

Jacketing of cylinder ends and pistons.—Steam-jackets should be designed to furnish a copious supply of heat to the surfaces in contact with the steam. Condensation is most abundant during the period of steam admission. Consequently the surfaces exposed at this time are those to which the application of heat is most beneficial. The combined exposed surface of piston and cylinder cover are equal to the surface of the cylinder bore, exposed when the piston has moved through a distance equal to one-half of its diameter. But during the first portion of the period of admission, in which condensation is most active, practically no part of the shell is exposed. Even after the lapse of 9 per cent. of the total time occupied in the stroke, the piston has only moved 2 per cent. of the total distance; at which moment the surface of bore exposed amounts only to 9 per cent. of the exposed surface of cover and piston, assuming the length of stroke to be double the diameter of cylinder. The most perfect jacketing of the shell only is therefore very far from providing an efficient application. In most cases the cylinder covers are unfortunately left unjacketed, though the operation presents no serious difficulty, and should always be undertaken as a necessary part of a jacket system. The treatment is, however, distinctly incomplete unless the piston and piston-rod are also jacketed. Some little difficulty is encountered in this operation, though it can be satisfactorily effected by supplying and withdrawing the steam and condensed water through the piston-rods or special sliding-pipes. Probably, however, in very few cases will any important gain be secured by jacketing the piston, having regard to the cost involved in the original provision and maintenance.

Value of high temperature in jackets.—A jacket should be supplied with saturated steam of higher temperature—and therefore of higher pressure—than the steam in the cylinder. This will not absolutely prevent the ebbing and flowing of heat to and from the surfaces of the cylinder in contact with the working steam, but will tend to confine such alternations to a smaller thickness according to the degree of efficiency with which the jacket performs its duty. Some cooling of the cylinder walls, however, still takes place during exhaust, and a corresponding amount of condensation during admission, but much less than in the absence of jackets. Re-evaporation during expansion is accelerated and may be completed before the end of the stroke, so that no re-evaporation during exhaust is possible. Consequently less cooling of the cylinder surfaces occurs during the period of exhaust than would otherwise be the case, and a further reduction is effected in the amount of condensation which takes place upon the cylinder surfaces during the period of admission.

Supply and condensation of steam in jackets.—The heat imparted to the working steam in a cylinder by means of a steam-jacket is liberated from the steam in the latter by reason of condensation, so as to set free its latent heat. As the jacket steam must suffer condensation in the act of yielding heat, it follows that the whole must be constantly in a saturated condition, and therefore that its temperature will depend upon its pressure. But the working efficiency of the jacket depends upon its temperature being at all times above that of the steam in the cylinder. The jacket steam is therefore best supplied by an auxiliary boiler, at a pressure above that existing in the main boiler. But practical difficulties usually prevent or oppose the adoption of a separate boiler, when proper precautions

should be adopted to provide an efficient supply from the main boiler. In the majority of high-class engines, an actual condensation of about 10 per cent. of steam is necessary to supply the balance of heat, which, together with that due to reduction of temperature during expansion of the entire charge of steam in the cylinder, amounts to 42·75 units per horse-power per minute =

$$\frac{33{,}000 \text{ foot-pounds per minute}}{772 = \text{mechanical equivalent of heat.}}$$

An inspection of the reports upon trials made upon jacketed engines will generally show that the most successful results have been achieved with a jacket supply of about 10 per cent. of the total weight of steam supplied to the engine, or say one-ninth part of the working steam supplied to the cylinder. In such cases, under ordinary conditions, the total amount of steam consumed may be reduced 10 per cent. A jacketed engine should therefore be so arranged that 10 per cent. of the total steam may be supplied to the jackets. The several jackets should be separately supplied with steam. The utmost possible pressure should be obtained in the high-pressure jacket, the steam to the others being regulated by reducing valves according to the results of experience. Each jacket should be provided with a good safety-valve, spring-loaded and tested, well within the structural strength of the jacket. The high-pressure jacket will actually take less steam than the later ones, but in a triple engine its pipe should be proportioned to pass 3 per cent. of the total steam consumed by the engine, and in a compound 5 per cent. The pressure in the high-pressure jacket may be well maintained by the use of efficiently-protected pipes,

giving a direct and separate communication from the boiler of such area that the velocity of steam shall not exceed one-quarter to one-third part of the mean velocity suitable for cylinder supply, as given in the chapter on pipes. Such pipes will be found to be larger than those usually adopted, and should be smooth, uniform, and free from abrupt bends. In the adjustment of the amount of steam supplied to each jacket, the results of observations upon the heat rejected in the air-pump discharge will be found especially useful when available. Steam which has been utilized in one or more cylinders is sometimes used for the jackets of later cylinders, and in some cases the steam supply of one or more cylinders is carried through the jacket of the same cylinder on its way to the valve-chest. These practices cannot, however, be of any appreciable service, but may lead to great loss; and when a case is quoted in which jackets have been found to be of doubtful service, inquiry should be made as to whether such result is due to the arrangements described.

Circulation of steam in jackets.—The circulation of steam in a steam-jacket is a matter of great importance. The positions of the supply and discharge branches should be so arranged that the steam cannot make a short circuit from one to the other and leave a considerable portion of the surface unaffected. Strengthening ribs are often required to support the different parts of the jacket; these may be so applied as to render good service in the distribution of steam. Even if not absolutely required for strength, they may be worthy of adoption as distributors. The general course of the steam in the jacket will be downwards, on account of the necessity for drainage, and to avoid interference of currents.

Necessity for removal of condensed water and air from jackets.—Accumulations of water in a jacket prevent the access of steam to the flooded surfaces, and assist the cooling of the cylinder, so that the jacket (partially or entirely) acts prejudicially instead of beneficially. Every drop of water should therefore be removed at once upon its formation. This may be effected by means of automatic steam-traps, each provided with a fine strainer, or by a closed circuit connecting with the water in the boiler. By the latter means the condensed water may be returned to the boiler without loss of heat, and economy promoted. But it is more likely than the former means to lead to accumulations of air in the jacket, which are nearly as prejudicial to the proper discharge of the duty of the jacket as are accumulations of water. When a closed circuit is adopted, the condensed water should be passed through a nozzle, surrounded by a glass tube, in such a manner that the stream of water may be seen as clearly as the oil in a sight-feed lubricator, and its quantity estimated from inspection, otherwise the working of the jackets is liable to become irregular. Air-cocks must be provided at all points where they can be of service. If these are found to require very frequent attention, they may be replaced by traps actuated by temperature. One or more cocks should be arranged to receive a pressure-gauge, and, in addition, each jacket should be provided with a pressure-gauge as a permanent fitting.

Protection against loss of heat by radiation.—The outside of a steam-jacket should be thoroughly protected by cleading against loss of heat. Otherwise the dissipated heat causes great loss and also discomfort to the attendants in the engine-room. The edges of flanges and small details, which in other cases are of minor

importance, are found to disperse large quantities of heat, by reason of the high temperature prevailing in a jacket.

Conditions under which greatest advantage is to be derived from the use of a jacket.—The engines whose economical record is most susceptible of improvement by means of steam-jackets are those working with a high rate of expansion in each cylinder, a great range of temperature in each cylinder, a low number of revolutions per minute, and whose consumption of steam exceeds 15 or 16 pounds indicated horse-power per hour. Probably, however, all large engines—except those supplied with superheated steam—would benefit by the application of jackets which conform to the conditions set forth. Jackets which fail to do so are much better omitted.

Steam-jackets successfully and unsuccessfully applied. The utility of steam-jackets applied to slow-working pumping-engines, and others in which steam is largely expanded, and in which a good supply of steam to the jackets is provided, has been repeatedly and clearly demonstrated, alike in the old Cornish practice, and in best modern practice. But as to the advantages to be derived from their adoption in connection with engines making a greater number of revolutions per minute, and in which the expansion is divided over several cylinders, much difference of opinion is held. Engines essentially similar have been largely and successfully adopted with jackets at sea; but comparatively few data are available as to the performance of stationary engines of the class referred to. The past or present existence of examples in which the foregoing conditions have been disregarded, or which were subject to structural defects, principally such as are liable to leakage at joints, largely account for the disfavour

with which jackets upon stationary engines are regarded. In all cases, engines provided with jackets should be slowly and carefully warmed before starting, or joints will most probably be started, resulting in a long stoppage.

Jacketing of receivers.—In Cowper's arrangement of compound engines, the intermediate pipe, which conveys steam from the high-pressure to the low-pressure cylinder, is enlarged and surrounded by a steam-jacket, by which means moisture condensed in the high-pressure cylinder is re-evaporated more or less completely, so that dry steam—or perhaps slightly superheated—is supplied to the low-pressure cylinder, and greater efficiency is secured. The performance of the high-pressure cylinder is unaffected. The action of the steam in the low-pressure cylinder is totally different from that under the influence of a cylinder jacket, and the advantage secured depends entirely upon a reduction in the liability of the steam to become condensed.

Compression as affecting condensation and re-evaporation.—Heat may also be imparted to the cylinder surfaces by means of compression in the cylinder, effected by closing the communication to exhaust, before the end of the return stroke. The steam remaining in the cylinder is then compressed, and the pressure rises until at the time of admission it may reach the pressure of the admitted steam. In this operation resistance is opposed to the motion of the piston, by which resistance mechanical work is consumed and its equivalent of heat produced. Such heat is undoubtedly useful in increasing the temperature of the surfaces in anticipation of the admission of steam, and thereby effecting an appreciable reduction in the amount of steam condensed during the period of admission; but the cost at which this is secured raises some doubt as to the economical results attending the practice.

Probably any benefit which is secured arises from the fact that the exhaust-port is closed earlier than it otherwise would be, and the cooling action of the exhaust upon the cylinder—effected by means of re-evaporation—is checked. The reason just given possesses greater force in connection with single cylinders, and the last or low-pressure cylinders of series. In each case, the absolute back-pressure is low, and compression commences at a comparatively early point in the return stroke. Compression is also adopted for mechanical reasons, apart from questions of economy.

Superheated steam used for the reduction of condensation.—Internal cylinder condensation, whether upon the surfaces exposed to steam, or effected throughout the mass by reason of the performance of work, is proportionately reduced by the adoption of superheating, whereby the steam is supplied to the engine at such a temperature as to admit of a considerable loss of heat before reaching the temperature of saturation at which condensation takes place. Neglecting losses by radiation and conduction, and in the total absence of condensation, 15 pounds of steam at an absolute pressure of 200 pounds per square inch would be required per indicated horse-power per hour, if superheated to 581·6° F., or 200° above the temperature of saturated steam at that pressure, and if it were expanded to 2 pounds absolute pressure, and a condition of exact saturation. In the practical use of superheated steam, a much smaller consumption is found sufficient, therefore some condensation must still prevail. Such condensation is, however, of much less amount than that which prevails in the absence of superheating, and the resulting moisture probably remains diffused in the mass of steam in such a manner that the surfaces are not appreciably affected. If under such conditions the

condensation were to reach an amount sufficient to produce 40 per cent. of the total indicated power, only 9 pounds of steam would be required. Such an amount of mass condensation would, however, assuredly lead to cylinder surface condensation and re-evaporation, and cause serious loss, so that the 9 pounds of steam would be exceeded. Condensation which does not affect the surfaces is beneficial, as it provides the means for the partial utilization of the latent heat of the steam. In an engine of any ordinary design, superheated steam is not liable to condensation during the period of admission, nor until it has been subjected to considerable cooling by the performance of work. Condensation, however, occurs when the temperature of the steam falls to the temperature of saturation corresponding to the pressure to which it is exposed at the time, and the abstraction of heat is continued. This is, however, never found to give rise to any important amount of surface condensation and re-evaporation, and special precautions against such action are therefore unnecessary. Steam-jackets are no longer necessary or desirable. If in any exceptional case there should appear to be any reason for the adoption of jackets, a supply of saturated steam at a temperature above that of the superheated steam should be adopted. The use of superheated steam in such jackets would be attended by irregularity of action, as it could not yield any of its latent heat except at a temperature much below that of the steam admitted to the cylinder.

Past and present practice with reference to superheating.—Twenty or thirty years ago superheating was very largely practised with success as regards the work done. The high temperature was, however, found to cause decomposition of the oil or tallow at that time used for lubricating the cylinders. The substances

left after such decomposition aggravated rather than diminished friction, and caused the mechanical destruction of the surfaces. The damage was also accelerated by the corrosive properties of the same substances at high temperatures. At the period in question, all piston-rods, valve-spindles, and other like details were provided with stuffing-boxes in which hemp, flax, cotton, or other vegetable packing was almost exclusively used. These were also rapidly destroyed by exposure to superheated steam. At the present time, however, mineral oils are available for the lubrication of cylinders, which oils will withstand in safety any temperature up to 600° F., beyond which it does not appear desirable to proceed. Such oils are exposed momentarily in gas-engines to a much higher temperature without causing serious difficulty. Asbestos packing, with or without the addition of other mineral substances, is largely used in stuffing-boxes exposed to all temperatures. The adoption of metallic packings for stuffing-boxes is a further step in the same direction. No difficulty of moment need therefore now be anticipated in connection with the cylinder, when adopting the principle of superheating steam.

Temperature of gases applied to superheater.—At the time referred to, steam was usually superheated by passage through a cylindrical or annular drum, or a tubular heater fitted into the chimney or flues and exposed to flue gases of a very high temperature, in consequence of which superheaters of all kinds adopted were remarkable for the rapidity with which they became worn or burnt out. In modern practice, all kinds of boilers are arranged to abstract from the gases a much larger proportion of the heat than was formerly abstracted. The gases are therefore discharged at a lower temperature, and are accordingly less liable to

cause damage to a superheater. Some superheaters are now heated by separate firing, and are said to be quite successful in work. Separate firing is, however, not to be recommended, as the heat must be more intense than in the spent gases from a boiler, and the heating surfaces of a superheater cannot impart heat to steam with the same facility as the surfaces of a boiler can impart heat to the water in the boiler. The waste gases, after passing through a well-proportioned economizer, are too far reduced in temperature to impart heat to a superheater. The superheater should therefore be placed between the boiler and the economizer, by which means also the coldest gases will be applied to the cold water, and the heat abstracted from the waste gases with the greatest degree of completeness.

Partial superheating.—Superheating is sometimes most successfully adopted for the simple drying of steam, or the complete conversion into vapour of all moisture which it may contain, but without proceeding sufficiently far to show any appreciable increase in the temperature of the whole. As in the adoption of steam-jacketing, a comparatively slight amount of superheating may prove to be of great service in keeping the surfaces dry, thereby restricting the interchange of heat to and from the cylinder, which interchange is much facilitated by the existence of a very thin film of moisture upon the surfaces.

Superheating in continental practice.—Superheating has been adopted during recent years much more extensively on the Continent than in England, and very successful results have been achieved. The consumption of steam in an ordinary unjacketed compound engine of high class may generally be reduced by about 15 per cent. upon the adoption of superheating, and the coal consumption a little more or less according to the conditions,

and to the type of superheater adopted. A saving twice as great is sometimes secured in the treatment of an engine, which from its small size or other reasons consumes an exceptionally large amount of steam for the work done.

Safe temperatures to which superheaters may be raised.—Steam may be superheated to a temperature of 600° F. without incurring any exceptional risk. Beyond this temperature the strength of wrought-iron rapidly diminishes as the temperature rises. The strength of steel is still more affected, and brass and copper are inadmissible. Wrought-iron is subject to loss of substance by corrosion at all temperatures, but such tendency is found to increase as the temperature rises. In dry air, at atmospheric temperatures, a piece of polished iron or steel may remain unaffected for a long time, but any considerable increase in temperature is at once followed by oxidation, which is immediately shown by the formation of a coloured film of oxide. On this account it is better that the steam should not be superheated beyond 550° F., at which temperature it is found that the principal advantages of superheating are secured. Aluminium would probably be found to resist oxidation with great success in the tubes of superheaters, for which it is admirably adapted, if it can be relied upon to resist molecular deterioration. In making a trial, it would be well to depend only upon the strength of the cast metal, and not upon any strength imparted by rolling or analogous processes.

Construction of superheaters.—All modern superheaters are made with a large number of tubes. These should expose sufficient surface for the reception of heat, so that not more than 1000 heat units are transmitted per square foot per hour when the temperature of the superheated steam is 200° below the mean temperature

of the furnace gases. If this difference reaches 500°, the heat transmitted may reach 1500 units, but this is not recommended, as the superheater may then suffer more severely from overheating, and unless additional heat is extracted by means of a feed-heater, such a difference in temperature must involve a loss in economy. In calculating the amount of heat absorbed by the superheater, the specific heat of steam under constant pressure may be taken at ·475. But if the steam should not be supplied to the superheaters in an absolutely dry condition, a corresponding amount of heat is absorbed in evaporating such moisture. Steam of 150 pounds pressure, containing 5 per cent. of unevaporated water, will, before showing any increase in temperature, absorb as much heat as would raise the temperature of the same weight of dry steam by 90° F. In this process the volume will be increased, nearly in the proportion $\frac{100}{95}$, while the pressure remains constant. The volume of dry steam is increased by superheating in proportion to the absolute temperature acquired—*i. e.* in proportion to the degrees on Fahrenheit's scale + 460. Thus, 1 cubic foot at 400° F. becomes 1·116 cubic feet at 500° F. $\left(\frac{500 + 460}{400 + 460} = 1{\cdot}116\right)$. In this case also the pressure remains constant throughout the operation. By this means the volume of steam is increased through the expenditure of a much smaller amount of heat than would be required for the production of an equal volume of saturated or wet steam, and at the same time the steam is in a condition more adapted for efficient use.

Circulation of steam in a superheater.—A superheater should be so arranged that the steam circulates freely over the heating surfaces. If the superheater is used for the extraction of heat from the waste gases, and these are not afterwards passed through an economizer for heating feed-water, it is desirable that the

superheater shall extract the utmost possible amount of heat from the gases. For the fulfilment of this condition, the course of the steam through the superheater should be in the opposite direction from that of the gases outside. But where an economizer is employed for subsequent treatment of the gases, the opposite course may be adopted. The incoming steam should then be distributed over the hottest tube-plate, and the life of the superheater prolonged. Assuming that the steam is most conveniently supplied and withdrawn at the opposite ends of the top of the superheater, there is some probability of the establishment of a short circuit, by which means the lower tubes may escape duty and suffer more damage by overheating. This may be prevented by the use of diaphragm plates extending from the top of the superheater to about the centre line, and at a sufficient distance from each tube-plate to clear the inlet and outlet branches. If desired, a third diaphragm may be applied in the centre of the length, reaching from the bottom of the superheater to about its centre.

Superheater fouled by wet, dirty steam.—Though an efficient superheater will deal with the wettest steam, and supply it in a perfectly dry condition, yet it is very important that it should receive the steam in the driest possible condition. This is on account of the solid matters carried over by moisture in the steam, which are deposited in the superheater, impairing the efficiency of the surfaces for the transmission of heat, and reducing the durability of the structure. Obviously this does not apply to water which has been condensed from the steam in transit through the pipes, but it is nevertheless important that such condensation should be minimized. Any solid matter will be deposited in the superheater, near to the point of supply, and if the

amount is large it may be necessary to revert to the arrangement in which the steam is supplied to the cooler end of the superheater.

Intermediate superheating.—Superheating may be applied to the intermediate pipes or receivers of compound triple or quadruple expansion engines. This may be conveniently effected by means of pipes traversing the steam-pipes or receivers, and exposing heating surface, the heat being supplied by live steam of higher temperature. As in steam-jacketing, the operation is attended with greatest benefit when applied to the last cylinder in a series.

Gehre's superheater.—Gehre's superheater has been used in some hundreds of cases on the Continent with great success, and is now supplied in England by Messrs. Donkin. In this arrangement, two cylindrical vessels are employed, usually placed side by side in the main flue leading from the boiler to the economizer or to the chimney, but not in a flue which contains the cool gases, after they have surrendered heat in the economizer. The superheaters are filled with tubes, somewhat after the manner of a tubular boiler, except that the tubes reach over the whole length and volume of the shells. The hot gases are able to circulate freely around the outside or through the tubes indifferently. Perfect access to the outside is facilitated by the fact that brickwork supports are dispensed with, the whole being suspended from above by light iron straps. Two vessels are adopted, so that thinner plates may be used for the shells than would be possible if only one were used, the total weight being approximately equal to that of one of equal sectional area and factor of safety.

Uhler's superheater.—Uhler's superheater is somewhat similar to a vertical boiler, with a number of hanging tubes suspended over a separate fire. Inner

tubes are also used in a manner similar to "Field" tubes for boilers. But the outer tubes which are closed at the bottom, and the inner tubes which are open at both ends, are at the upper ends connected to separate boxes, one of which is the supply-box and the other the collecting-box. By this means a positive circulation is maintained, so long as the joints and tubes are good. The steam will either pass down the outer tubes and up the inner ones, or in the reverse direction. Hitherto the weakest point of this apparatus consists in the liability to burning of tubes, owing to the deposit of solid matter from the steam, though they are set eight feet above the fire.

Superheaters of continuous tubular construction.—The high-pressure boiler of Perkins, designed about thirty years ago, and the modern one of Serpollet, are well adapted for superheating. But owing to the inaccessible character of the tubes, they should not be used for this purpose, except in conjunction with vessels of some kind, in which to perform the first stage of superheating, and to effect the deposit of any solid matter. Such primary vessels should obviously be arranged for durability and accessibility for cleaning.

McPhail and Simpson's superheater.—This superheater has been extensively applied in England recently. It consists essentially of a number of serpentine tubes, placed in the back flue, and exposed to the hot gases as they leave the internal flues of the boiler. The battery of superheating tubes is connected with pipes, placed in the water space of the boiler, and so arranged that the temperature or degree of superheat in the steam may be reduced in any required proportion, the excess of heat being imparted to the water in the boiler, or practically increasing the heating surface.

Furnace superheaters.—Superheaters would be more

efficient in proportion to their dimensions and cost if placed in the furnace-tubes of the boiler. These would work well under ordinary conditions. But if at any time the engine should stop suddenly, so that the current of steam should be arrested, the superheaters would be rapidly destroyed, as the temperature of one cubic foot of superheated steam would rise 400 times as much as that of one cubic foot of water by the access of a given amount of heat. At all times, also, the wear and tear would be liable to become excessive.

Economy secured by superheating.—The use of well-designed superheaters is attended with saving in all cases, and may be adopted in the fullest confidence that the troubles of twenty or thirty years ago will be easily avoided. The greatest percentage of saving is, however, usually secured in cases where the engine or its appointments are most defective, where the weight of steam consumed is greatest, and especially where a high rate of expansion is adopted without adequate provision against cylinder condensation. Also in small engines, in which large quantities of steam are usually consumed for the work done. A boiler will evaporate nearly the same amount of water per pound of fuel, whether the steam is subjected to superheating or otherwise. But the advantage secured in the engine will often lead to such a reduction in the total amount of steam required, that the boiler will be relieved from forcing, and by reason of more easy work will give a better evaporative duty per pound of fuel than before.

Fall of pressure, during expansion of superheated steam. From the absence of moisture in superheated steam, it follows that the pressure after the point of cut-off may be expected to fall more rapidly than when saturated steam is used. This is because the heat consumed in the performance of work must be withdrawn from

the sensible heat of the steam, unless or until the point of saturation is reached.

Excessive cylinder condensation generally due to high expansion.—When steam is admitted to a cylinder throughout a large proportion of the stroke, the period of expansion is correspondingly curtailed. The fall of temperature during expansion is also reduced, so that the cooling action by re-evaporation during expansion is practically avoided. The cooling during exhaust, by reason of re-evaporation, does not extend so far into the metal, less heat is carried away by the exhaust, and less condensation arises during the subsequent admission of steam for the succeeding stroke.

Conditions under which the use of a condenser may be dispensed with.—In the foregoing treatment of this question it has been assumed that a condensing engine is employed. In locomotives it has been found hitherto impossible to resort to the use of a condenser, and moreover the exhaust steam is utilized for the production of an exceedingly powerful chimney draft, without which the amount of steam produced would fall far below the point necessary for the work. For this purpose no means are found to be practically applicable except the blast-pipe supplied by exhaust steam. In very small stationary engines the use of a condenser and air-pump is generally considered to be unprofitable, unless the boiler should be insufficiently powerful to furnish steam without them. In larger engines used intermittently, as in centrifugal pumping, the same condition exists. But in connection with stationary engines worked during ordinary time and developing more than 50 indicated horse-power, condensation should almost always be adopted.

Range of temperature in a single cylinder increased by the use of a condenser.—The omission of condensation,

however, does not affect any of the conditions laid down, with the exception that if the same range of pressure be adopted as in a condensing engine the range of temperature is reduced. Assuming that one engine works at a pressure during admission of 150 pounds above the atmosphere, and a vacuum of 12 pounds below atmospheric pressure, the range of temperature in the engine—whether in one cylinder or in several successive cylinders—will be 224° F. A second engine may work without condensation from a pressure of 165 pounds above the atmosphere to a back pressure of three pounds above the atmosphere, which will give a temperature range of 158° F. The work done will, however, be very much smaller than in the former case, because while it gains an additional pressure of 15 pounds at the upper end of the scale, it loses an equal amount of pressure at the lower end of the scale, where the volume of steam is many times greater than at the higher pressure. In the cylinders of non-condensing engines the diminished range of temperature leads to a corresponding reduction of condensation and re-evaporation. But though on a reduced scale, these actions still proceed, and may be met by the same measures as in the cylinders of condensing engines.

Reference may be made to *Minutes of Proceedings of the Institute of Civil Engineers*, vol. c., p. 347, also vol. cvi., p. 264, for papers on temperatures of cylinder walls. Also vol. cxv., p. 263, for paper on cylinder condensation; all by Mr. Bryan Donkin. In the *Minutes of Proceedings of the Institute of Mechanical Engineers*, Oct. 1892, a report from the Research Committee on steam-jackets is published, with table of results.

CHAPTER XIX.

CLEARANCE AND COMPRESSION IN STEAM CYLINDERS.

Necessity for clearance.—In the cylinder of an ideal engine, the piston, when at the end of its stroke, would just touch the whole of the surface of the cylinder-cover, inclusive of any bosses, screw-heads, nuts, or other projections upon either. The steam-ports or passages would also be so arranged that, at such a time, no space would exist between the piston and the valves which control the entry or withdrawal of steam. If such conditions were possible, the first particle of steam admitted would tend to cause the motion of the piston, and thus perform useful work. But the necessity for thoroughfare area sufficient for passing the requisite quantity of steam in the time allowed, the provision for small disarrangements of adjustment in length of piston-rod by reason of wear, free compression of steam, and safety in the presence of a small quantity of water in the cylinder, combine to render necessary a large amount of clearance space, which space must be filled with steam before any work can be done by the steam.

Variation in clearance by length of stroke.—The total amount of clearance space is usually considered together and stated in cubic inches, or in a percentage of the volume swept by the piston, which latter is the product of the area of the piston multiplied by the length of its

stroke. In either case, the clearance thus given only refers to one end of the cylinder. The total amount may, however, for the present purpose, be divided into the cylinder clearance space and the port clearance space, the former being included between the piston and cover, while the latter is included between the main cavity of the cylinder, and the valve or valves controlling the steam. In carefully-designed engines the proportion of clearance varies with the length of stroke, the number of revolutions or speed of piston, and with the estimated necessity for ensuring free admission and exhaust of steam, so as to avoid excessive resistance in either operation; also with the views of the designer as to the objects set forth in the last paragraph. As a rule, the amount varies from 5 to 10 per cent., but may fall to 3 or rise to 15 per cent. Two engines whose cylinders are of equal bore, and which work at the same piston speed, require equal ports, so as to pass the steam with equal facility. The cylinder clearance space will also be nearly equal in the two cases. Consequently, the percentages of total clearance in the two engines will compare almost inversely as the length of the stroke. When steam enters the space behind the packing-rings in the piston, such space must be added to the clearance volume.

Steam consumption and work done as affected by clearance.—Two engines may be compared whose percentages of total clearance are different, but whose dimensions, piston speed, and steam-pressure are identical, and which cut off at the same point in the stroke. In such engines the amount of steam consumed will vary according to

(percentage of clearance + percentage of cut-off),

so that the engine with larger clearance will consume

more steam by the absolute difference of clearance. Assuming that the steam in each case is cut off at 35 per cent. of the stroke, and that the clearances are respectively 5 and 10 per cent., the proportionate amounts of steam consumed will be (35+5) : (35+10), or a difference of 12½ per cent. greater in the latter case. But the total amount of steam expands together, and thus the pressure existing in the cylinder at any

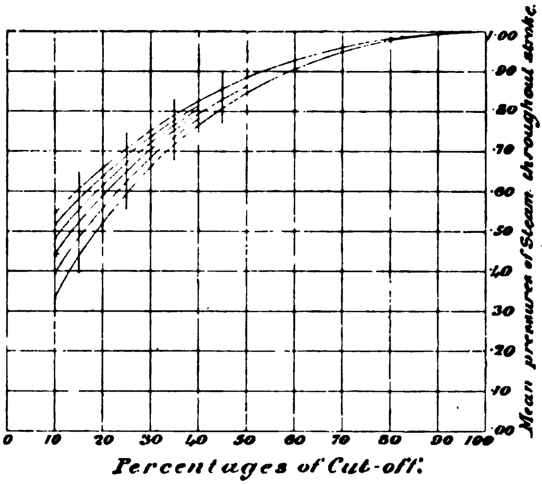

Fig. 25.—Mean pressures in cylinder, with various expansions and clearances.

moment during expansion after cut-off is increased by reason of the steam admitted to fill the clearance space. The average pressure throughout the stroke is therefore increased in this way, and more work will be done, though very much less than in proportion to the amount of steam used. Fig. 25 shows the comparative mean pressures throughout the stroke with different degrees of cut-off, and with clearances of 0, 5, 10, 15, 20, and 25 per cent., the last two being beyond the limits of ordi-

nary work. The lowest curve refers to no clearance, and the highest to 25 per cent. True hyperbolic expansion is assumed throughout, and the scale of ordinates is such that 1·00 denotes the initial pressure of the steam. Fig. 26 shows the comparative efficiency of steam, making allowance for the amount required to fill the clearance space, and for the increased mean pressure consequent upon the expansion of the steam contained in the clearance space. The vertical scale of this diagram is such that 100 parts denote the efficiency of steam admitted throughout the stroke (*i. e.* without expansion) into a cylinder without clearance. It will be seen that the loss by reason of clearance is greatest when the steam is cut off early. Thus, at a cut-off of 20 per cent. with a clearance of 5 per cent., the efficiency is 223, while at the same cut-off but with a clearance of 15 per cent. the efficiency falls to 176, or less than before, in the ratio 100 : 79. With a later cut-off the efficiency varies more nearly in accordance with the percentage of clearance.

Compound working, as affecting loss by clearance.—In a compound or triple expansion engine, the same steam, unchanged in amount, is dealt with successively in the several cylinders. As a rule, the percentage of clearance in each cylinder of the same engine is approximately the same, so that at each stage the same degree of loss results. The work done in each cylinder, and the sum of these, will therefore be each subject to the loss corresponding to the actual cut-off, and not to the total amount of expansion effected in the combination of cylinders. For example, two compound engines may be considered, the area of whose cylinders bear the ratio of 1 : 3, and in each of which the steam is cut off at 35 per cent. of the stroke. Each may also be compared with a simple engine in which the steam is

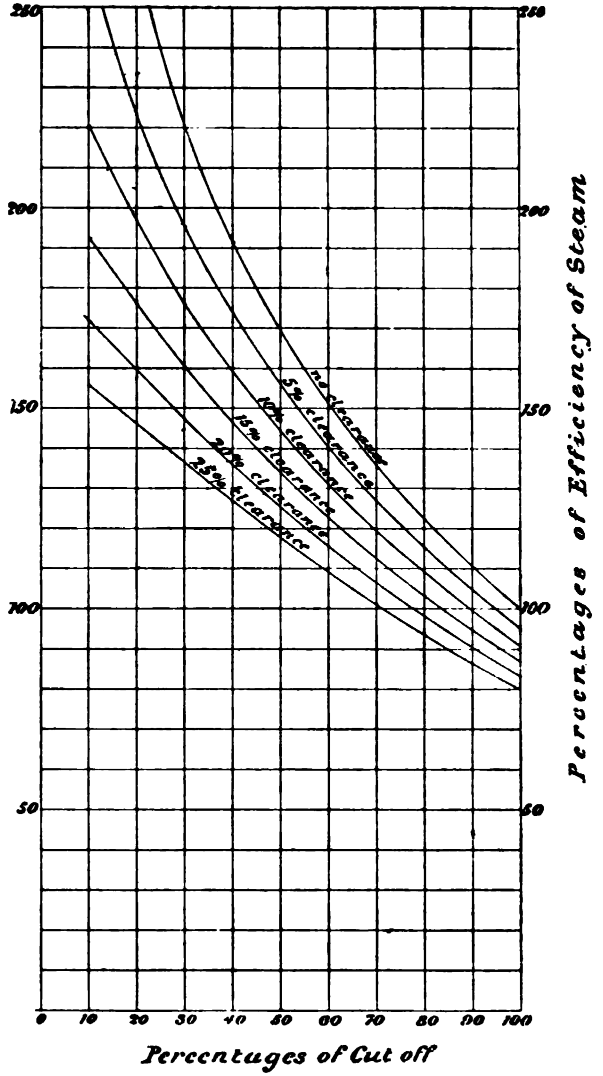

Fig. 26.—Efficiency of steam in cylinder, with various expansions and clearances.

expanded to the same degree, whose single cylinder is of dimensions equal to those of the low-pressure cylinder of the corresponding compound engine, and in which the same amount of steam would perform the same amount of work, except as affected by cylinder condensation, access or withdrawal of heat, leakage of steam, or clearance. Each may also be compared with the ideal case in which clearance is absent. The comparative results of such calculations are given in Table XII.

Compression.—In no case is the steam in a cylinder entirely withdrawn on the opening of the valve to exhaust. The weight remaining at any time may be found from the volume, and the weight per cubic foot corresponding to the absolute pressure at the moment. If at any time during the period of exhaust, and before the end of the stroke, the valve be closed to stop the escape of steam from the cylinder, the steam thus imprisoned, being compressed into a decreasing space, must be subjected to an increasing pressure. If this action be accurately adjusted, the pressure obtained in the cylinder may just reach that of the steam admitted. The clearance space being thus already filled before the opening of the steam admission-valve, the first particle of steam admitted is in a position to perform work, and the amount of steam required to fill the clearance space is saved. This action, however, causes the consumption of power, by which the useful work of the engine is reduced. The nett economical result to be secured by the adoption of compression varies largely under different conditions, and can only be ascertained by means of intricate calculations, in which suitable allowances are made on account of cylinder condensation. Table XIII. gives a comparison between the two engines treated in Table XII., on the assumption that the clearances in

TABLE XII.—COMPARISON OF SIMPLE AND COMPOUND ENGINES OF EQUAL NOMINAL EXPANSION, BUT WHOSE CLEARANCE VOLUMES VARY.

	Clearance.		
	0.	Five per cent. upon the volume of each cylinder.	Fifteen per cent. upon the volume of each cylinder.
Simple engines, cut-off at 11·67 per cent. :—			
Proportionate amount of steam used	100	143	229
do. do. work done	100	115	138
Proportionate amount of steam required for equal work	100	124	166
Compound engines, cylinders 1 : 3, *cut-off at* 35 *per cent.* :—			
Proportionate amount of steam used	100	114	143
do. do. work done	100	103	109
Proportionate amount of steam required for equal work	100	111	131

each cylinder amount to 5 per cent. upon the volume of the cylinder, that there is no loss of pressure at any point, and that the back-pressure is 5 per cent. of the steam-pressure. When modified to suit working conditions, the absolute figures would be different, but their general relation to each other would be the same. In the

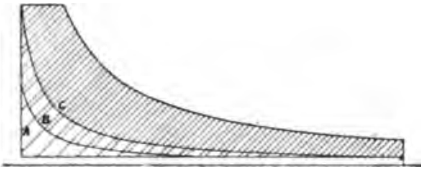

Fig. 27.—Diagram relating to simple engine treated in Table XIII.

simple engine, the amount of steam required for the work done is less when half compression is adopted than it is either when full compression is adopted or when compression is avoided. Possibly further trial would show a different degree of compression, which would give still better results. In the compound engine, the loss by clearance has been shown to be much less

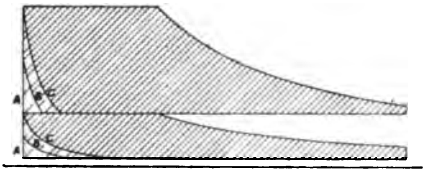

Fig. 28.—Diagram relating to compound engine treated in Table XIII.

than in the simple engine. But the mechanical work of compression is performed with greater efficiency in stages than in one continuous operation, and as a consequence the loss by reason of clearance is more completely recovered than is possible in the simple engine. Figs. 27 and 28 show the diagrams from which the results given in Table XIII. have been obtained.

TABLE XIII.—COMPARISON OF SIMPLE AND COMPOUND ENGINES, WORKING UNDER DIFFERENT DEGREES OF COMPRESSION, OF EQUAL NOMINAL EXPANSION; CLEARANCES EQUAL TO FIVE PER CENT. UPON THE VOLUME OF THE RESPECTIVE CYLINDERS.

	Degree of compression.		
	No compression. Figures transferred from Table XII.	Sufficient to give pressure in cylinder, half-way between back pressure and initial pressure.	Sufficient to raise pressure in cylinder to equal initial pressure.
Simple engines, cut-off at 11·67 *per cent.* :—			
Compression lines in Fig. 27 ...	*A.*	*B.*	*C.*
Proportionate amount of steam used	143	122	100
Proportionate amount of useful work performed	115	104	83
Proportionate amount of steam required for equal work	124	117	120
Compound engines, cylinder areas, 1 : 3, *cut-off at* 35 *per cent.* :—			
Compression lines in Fig. 28 ...	*A.*	*B.*	*C.*
Proportionate amount of steam used	114	107	100
Proportionate amount of useful work performed	103	101	96
Proportionate amount of steam required for equal work	111	106	104

CHAPTER XX.

TYPES OF STEAM-ENGINES.

Horizontal engines.—In current practice stationary steam-engines are more frequently placed in a horizontal position than otherwise. The chief advantage of this arrangement is the superior degree of accessibility to all glands, covers, valve-gear, and other parts which is secured. As generally constructed up to a recent period, the whole of the cylinders, slide-bars, valve-gear, and crank pedestals were separately placed upon a flat framing or bed-plate, which arrangement presents many points of convenience in manufacture. In all such cases, and especially if the cylinders are secured by feet placed considerably below the central plane, bending strains arise in work which act alternately in opposite directions. These must be met by a sufficient quantity of material in the bed-plate, and by a considerable depth, or the whole will move appreciably at each stroke. This defect is often avoided in more recent work by the adoption of the system of framing in which the working stresses are directly opposed, and which is elsewhere described. A horizontal engine presents the maximum facility for adjustment of valve-gear, slide-bars, cross-heads, and connecting-rods. A slight complication exists in connection with the crank-shaft bearings,

which are subjected to the stress of the connecting-rods in a horizontal direction, and to the weight of the fly-wheel acting in a vertical direction. The latter is, however, in many cases so very much greater in amount than the former, that no serious trouble arises in this way. In some cases, however, the crank-shaft bearings are not equally loaded, when the horizontal movement of the shaft must be prevented by means which admit of fine adjustment, but provide rigid support and ample surface of bearing against the shaft; such facility is also very valuable even when not imperatively necessary. The wearing surfaces, in connection with which there has really been the most serious difficulty experienced, are the bores of the cylinders and the lower edges of the pistons. These have sometimes given trouble, but with suitable proportions, and the use of mineral oils, administered by means of sight-feed lubricators, such difficulty is rapidly disappearing. Also by cambering the piston-rods to an accurately-adjusted amount, the pistons may be caused to bear against the top or bottom of the cylinders, or run just free, when each end of each rod is truly supported by slides.

Length of stroke in horizontal engines.—Long strokes possess undoubted advantages, which have perhaps been over-estimated. These have often decided the adoption of horizontal engines as the only type to which in the particular cases long strokes could be applied. Great length of stroke involves a low speed of revolution. This necessitates a great power of fly-wheel, which may be obtained either by weight or diameter. In a horizontal engine there is usually but little restriction upon the diameter, so that the weight need not be excessive. In other types, the diameter of fly-wheel is often limited, so that the power required may involve a great weight of wheel. But the greater speed

of revolution which follows from the shorter strokes usually adopted tends to reduce or neutralize such disadvantage.

Space occupied by engines.—Horizontal engines occupy much greater floor space than vertical ones, but less height. In this way the dimensions of space available for an engine will often determine the type to be adopted in a particular case. Owing to the large amount of floor space occupied, the cost of the foundations and of the house for a horizontal engine is comparatively great.

Accommodation for pumps.—The pumps of horizontal engines are generally placed beneath the floor of the engine-house, and worked by a lever or L leg. This may be driven from the crank-pin by a separate connecting-rod, or from the cross-head pin by means of a pair of links. In rare instances, in which a parallel motion is employed, the pumps are worked from this. In the former cases the pumps are placed in the foundation, in positions very difficult of access, and are very apt to be neglected. When a parallel motion is adopted there is much less difficulty in placing the pumps in a position of greater accessibility, which is an advantage in every respect. Sometimes a horizontal air-pump is adopted, worked from a continuation of the piston-rod, and placed on the engine-room floor. This is perfectly accessible, and may be made quite efficient; but for various minor reasons such air-pumps have not yet met with great favour.

Vertical inverted engines.—The vertical inverted type of engine is the only one which practically competes with the horizontal one. Above the level of the cross-heads, all motion and strain follow a strictly vertical direction, so that no trace of a necessity for side support exists. The whole force due to the pressure of steam

upon the piston and cylinder-covers is imposed upon the standards in such a manner that these are alternately subjected to extension and compression,—when the engine is at work,—and caused to vary in length. The standards should be so designed that all parts situated upon the same horizontal section shall be exposed to the same stress per square inch. Otherwise one part will suffer extension or compression to a greater extent than another, and the cylinder will be thrown out of line. In this way a very small variation of length will give rise to a most objectionable amount of vibration, of a character which scarcely any practicable system of staying will overcome. In the best and simplest arrangement of stationary engines, the standards are of cast-iron, of ample proportions, arranged with scrupulous regard to symmetry, with respect to the vertical plane which traverses the axes of the crank-shaft and piston-rod; and also with respect to the vertical plane, which is at right angles to the first, and which also traverses the piston-rod. Fifteen years ago this condition was often disregarded in marine engines. The front standards were made of light section of wrought-iron, so that they suffered greater extension than the back standards, and a steady engine was almost unknown. But in current practice the matter receives great attention, and the engines are found to benefit largely by the change. In mercantile marine practice, the back standards are usually worked into the condenser casting, so that some disparity exists; but the front standards are made of large section, so that the absolute variation in length becomes exceedingly small.

Defective example.—The most unsteady engine which ever came under the notice of the author was a vertical inverted one, in which two cylinders were placed near to each other and mounted on the top of one double

standard. One cylinder overhung each side, so that the line of the piston-rod fell outside the standard. This engine might have worked with fair steadiness with the pistons in unison, and at all times equally loaded. But in the actual case, the cranks were set opposite to each other, so that the effect of the whole arrangement was to secure the greatest possible bending stress upon the standard, precisely as though the utmost possible degree of vibration was sought to be secured.

Advantages of vertical engines.—In well-designed vertical engines, the crank-shaft, connecting-rods, and valve-gear are conveniently accessible. All glands and cross-head slides are very accessible and easily adjusted. The side-thrust from the cross-head blocks is not apt to cause any serious amount of deflection, as it is applied at a low level, and is of comparatively small amount. The whole of the strains upon the crank-shaft are applied in an approximately vertical direction, with the small exception of that due to the taking of the work from the fly-wheel. The wear of the brasses is therefore in a vertical direction, and is most easily dealt with. The weight of the piston is also imposed upon the cross-head pin, crank-pin, and the crank-shaft, in each of which it is most efficiently resisted, and lubrication can be applied.

The pumps of vertical inverted engines are usually worked by means of levers driven from the cross-heads. The pumps may be placed at any level, but are usually above the engine-room floor and most freely accessible.

Foundations.—The cost of foundations necessary for a vertical engine is in any ordinary case much less than for an equivalent horizontal engine. As a rule, the cost of the house will be found to compare in the same way.

Uniformity of motion.—As to regularity of turning, the vertical inverted engine gives a better result than

a horizontal one. This advantage is also increased by reason of the high speed of revolution usually adopted in combination with the short stroke of such engines. The efficiency of a fly-wheel as a regulator of speed may be practically equal in each type of engine, having regard to the several factors dealt with in the chapter on fly-wheels.

Attention to cleanliness.—A vertical inverted engine will rapidly degenerate into a dirty, bespattered condition if neglected. In this respect, however, a very little patient attention will suffice to reveal the source of any drip or spray which may arise from the glands, cross-heads, or cranks, and to suggest measures for its prevention.

Vertical engine with triangular connecting-rod.—A special modification of the vertical inverted engine has been lately introduced to stationary work by Messrs. Musgrave of Bolton, in which two or three piston-rods are attached to a connecting-rod of triangular profile. This arrangement is especially fitted for the application of the quadruple expansion principle to two cranks, or for compound or triple expansion to an engine with only one crank, either engine occupying a minimum space. This system has also been successfully applied to engines of small size, designed to work at a high speed for driving dynamos, and for other equivalent work.

Beam-engines.—Beam-engines are now very seldom adopted in new works, though occasionally so in cases of extension, so as to harmonize with existing works. This type of engine is mechanically a most efficient one, and all parts are freely accessible. The strains to be resisted, however, act at many points, and in a great measure independently of each other, so that the cost is comparatively great. An additional reason for a high

cost consists in the fact that the system does not lend itself to the adoption of a high speed of revolution, consequently an engine of large dimensions is necessary. The pumps are directly worked, and the action of the parallel motion is much superior to that of cross-head slides. The high cost of a beam-engine, the foundations, and the house for its accommodation, are the present obstacles to its extended use. The beam is frequently—but not necessarily—a weak point, causing much trouble and expense by breaking, and in many cases leading to a complete collapse.

Compound beam-engines.—Beam-engines were originally built with one cylinder, connected to one end of the vibrating beam, and the connecting-rod attached to the opposite end, when the whole of the work was transmitted through the whole length of the beam. In almost all cases, however, a second cylinder is applied to the same beam for working in compound circuit with the main cylinder. In Woolf's system, the second cylinder is applied to the same end of the beam, as close to the point of application of the first one as is convenient, and is used as the high-pressure cylinder. In this case the pressure upon the centre is rather greater than with a single cylinder of power equal to the two Woolf cylinders. In MacNaught's system, the second cylinder is also used as the high-pressure one, the main cylinder taking steam from it. The usual position for the centre line of the high-pressure cylinder is mid-way between the centre of the beam and that of the connecting-rod. As in other cases, the total amount of power produced by the engine is approximately equally divided between the two cylinders. When this is the case in a MacNaught engine, the pressure upon the centre due to the work done is only one-quarter of that in a simple engine. By placing the high-pressure

cylinder nearer to the centre of the beam the pressure is reduced, so that at one-third of the half-length of the beam the two pressures are accurately compensated, and the centre of the beam is exposed to a pressure due only to the weight of the beam and details suspended from it. As affecting the stresses upon the beam itself, the several cases may be treated on the principle of the lever. But the load upon the centre affects the structure which supports the bearings.

Vertical overhead-crank engines.—Vertical engines with overhead cranks are practically extinct. Their original cost was not excessive, but they were inconvenient to work, and the crank-shaft was difficult to control. The pumps were usually worked by means of levers, and were generally—though not necessarily—very inaccessible. If this type should ever be restored to favour, it will probably take the form of a self-contained engine, with a crank-shaft mounted on two massive "A" frames, arranged to stand on a bed-plate which will also support the cylinders above the floor. The air-pumps should be placed at as low a level as possible, and may be most efficiently worked by rope pulleys or levers. A point of great importance will be as to the sufficiency of material in the A-frames, and the manner of its distribution, so as to secure uniformity of extension in the several parts, or to reduce its amount to a minimum. A second point, of but little less importance, is the balancing of the crank and connecting-rod end. These remarks apply with equal force to diagonal or inclined engines, which are seldom adopted except for special reasons. In many such cases two cylinders are connected to one crank, which arrangement is a complicated one; cramped proportions are often adopted, as a result of which the work is very difficult to keep tight and free from knock at the crank-pin. The last point

may appear to be a trivial one, but experience shows it to be far otherwise. There is, however, but little probability that overhead-crank engines will be largely adopted, whether with vertical or inclined cylinders, even as compounds, but still less as triple expansion engines.

Tandem engines.—Many horizontal engines, and some vertical engines, are made with two cylinders to work in compound circuit, and with pistons attached to the same piston-rod. This class is known as a "tandem" engine, and may be adopted either with a single crank or with two cranks, in which latter case each side is practically a separate engine, and the cranks are set at right angles with a view to uniformity of turning. The steam from one high-pressure cylinder is generally confined to its own low-pressure cylinder. If the high-pressure exhaust-valve and the low-pressure admission-valve open and close simultaneously, the exhaust-pressure upon the high-pressure piston will correspond with the steam-pressure upon the low-pressure piston, subject only to correction on account of resistance encountered in the ports and pipes, and the expansion will be almost identical with that obtaining in a single cylinder, but with diminished condensation due to the adoption of compound working. Some pressure may be lost in this operation on account of excessive volume in the intermediate-pipe and low-pressure valve-chest, which requires to be re-filled at each stroke. Such loss of pressure will, however, equally affect the pressure in each cylinder. In a well-designed engine, such correspondence in pressure actually prevails during the period of steam admission to the low-pressure cylinder, after which the closing of the low-pressure admission-valve causes compression of the steam remaining in the high-pressure cylinder

and the intermediate-pipe, due to the reduction of volume by means of the high-pressure piston. If the amount of this space is accurately adjusted, the pressure may be caused to rise very near to the pressure of release in the high-pressure cylinder, so that the drop in pressure is minimized. A similar condition prevails in a Woolf engine, and in the few engines in which two cylinders are placed parallel and near to each other, and are connected to cranks placed in the same plane. The case of a MacNaught engine may be slightly modified by reason of a longer intermediate-pipe.

Order of cranks upon crank-shaft.—A very large number of engines are made in which the successive cylinders are connected to cranks which are not in the same plane. In most cases the cranks are at right angles to each other, but in others are set at different angles. In such cases it is clearly impossible to arrange that the point of admission for one cylinder shall coincide with that of release from the preceding cylinder. Consequently, some amount of receiver space is required between the two cylinders, with a view to regulation of the pressure. The best position for the crank of the second cylinder is about 45° in advance of the position directly opposite to the first one, or is 135° behind the first. When the first piston is at the end of the stroke, the second piston will have traversed about 14·6 per cent. of its stroke. By this means a more uniform motion is secured, but the amount of power produced by the steam may be nearly the same. In many triple engines three separate cranks are adopted for the respective cylinders. These are usually placed at intervals of 120° apart. Each should be so placed that they will reach the top of the stroke in the order—high, intermediate, and low pressure.

Steam consumption of different types of engines.—

Simple engines in which the expansion of steam is completed in one cylinder are practically never adopted in new work, on account of their extravagant consumption of steam. High-class engines of this type, with an efficient cut-off, consume about 21 pounds of steam per indicated horse-power per hour. Others, apparently but little inferior, will take 25, 30, or 40 pounds of steam. Compound engines in which the steam is expanded in two cylinders will take about 17 to 20 pounds of dry, saturated steam. Triple expansion engines working up to 160 pounds pressure will consume about 13 pounds of steam per indicated horse-power per hour, and quadruples somewhat less. In all cases a saving of from 10 to 20 per cent. may be anticipated to follow the adoption of superheating. In each case a thoroughly well-designed engine of its class, free from complications, and which can be soundly constructed and worked at a reasonable cost, is assumed. Every detail must be in order to secure the low consumption of steam stated. On the other hand, work of an exceptionally high class will secure better results.

CHAPTER XXI.

ADJUSTMENT OF PROPORTIONS OF CYLINDERS OF STEAM-ENGINES.

Estimates of power required.—In deciding upon the proportions to be adopted in an engine, the first element to be ascertained is the amount of power which is required to be furnished. In some cases this is obtainable separately, for the machinery to be driven, and for the shafting and mill-work in connection, upon which a close approximation may be made as to the additional power required to drive the engine. Usually, however, it is only possible to make a rough estimate of the total power required, which is often based upon the results obtained, or supposed to be obtained, elsewhere, under different conditions. In many cases only a conjectural estimate can be made, and as a result great loss is often incurred, either in providing and working an engine much too large for the work, or in loss of time by reason of deficiency in power. In connection with estimates as to power, every item should be checked, and the possibility of changes in process, or in description of goods manufactured, or in speed of machinery, should be considered. Also possible extensions of plant, and any other question to affect the power which the engines may at any time

be called upon to furnish. When an important amount of variation is to be anticipated, arrangements should be made whereby the engine will work to the best advantage during the longest time. Provision for future additions may be made by arranging for the addition of one or more cylinders; or by originally fitting liners in the cylinders, to be removed when more power is required. As a rule, the duplication of cylinders is conducive to steady turning, but except when applied to increase the stages of expansive working, and under suitable conditions for the purpose, there is likely to be some loss in economy. Probably the majority of engines devoted to miscellaneous work are much larger than is necessary, and a very small amount of attention devoted to the question in the first instance would be amply repaid.

Pressure of steam.—For the sake of economy, the highest practicable pressure of steam should be adopted. In special cases, however, comparatively low pressures are wisely adopted. The chief reason for this step is the desirability of utilizing existing boilers for the production of steam. Another is that only a small power is required, and consequently there is only a small item of working cost upon which a saving can be made, while the more perfect appliances necessary to economical working are costly to provide in the first instance. These reasons apply chiefly to engines used for temporary purposes, or in permanent service, which is only required intermittently, and in connection with which there is no deficiency of boiler power.

Estimation of horse-power of engine.—The power which a single-cylinder engine may be expected to develop may be readily estimated by taking the difference between the mean steam-pressure and the mean back-pressure in the cylinder, in pounds per

square inch, which multiplied by the nett area of the piston in square inches, and by the piston speed in feet per minute, gives the work done by the engine in foot-pounds per minute. This quantity, divided by 33,000, gives the indicated horse-power. When steam is admitted throughout the stroke, the pressure against the piston is practically uniform. But when steam is cut off at an early part of the stroke the pressure falls, whereby the average pressure is reduced. Table XV. gives the several ratios which apply to different degrees of cut-off. These are corrected for different degrees of clearance, and are based upon the assumptions that the pressure is uniform throughout the period of admission up to the point of cut-off, and that the pressure during expansion falls in strict accordance with the hyperbolic or isothermal law. The mean steam-pressure is found by multiplying the pressure at cut-off by the proper ratio selected from the table. If the pressure at cut-off is 75 pounds, cut-off at 40 per cent., and the clearance 10 per cent., the mean steam-pressure $= 75 \times \cdot 794 = 59 \cdot 6$ pounds. This is not the mean *effective* pressure, but requires to be reduced by the amount of the back-pressure before it can be used. In all calculations, the pressure of steam must be measured from zero, so that all pressures stated in the ordinary way—in pounds per square inch above the atmosphere—require to be increased before adoption. The amount of increase should exactly correspond to the pressure of the atmosphere at the time and place, but for most practical purposes may be taken with sufficient accuracy at 14·7.

Successive pressures during expansion.—According to the law of hyperbolic or isothermal expansion, the pressure exerted by a given quantity of a gas confined in a vessel varies inversely as the volume, so that the

product of the volume and pressure multiplied together is constant, however much each of these attributes may vary. This is strictly true of perfect gases, when expanded without performing work, or suffering any change in temperature. An inspection of Tables IX., X., and XI. will, however, show that the weight of saturated steam per cubic foot varies slightly from this condition. Steam suffers cooling by reason of expansion, by which, considered alone, its volume is reduced. Cylinder condensation of both kinds and re-evaporation also cause departures from the true hyperbolic curve of expansion. The amounts of such departures are, however, so moderate while the engine is in good condition, that the true hyperbolic curve is still a convenient standard of comparison for the expansion curves of indicator diagrams. Condensation which occurs during the period of expansion causes the curve of expansion to fall with greater rapidity, while re-evaporation has an opposite effect. The effect of clearance is to raise the curve, or rather to assimilate it with the curve which would be produced in the absence of clearance by delayed cut-off.

Location and importance of point of cut-off.—In all cases of calculation as to the power of proposed engines, or in trials, or in comparisons of diagrams, the actual point of cut-off is of great importance. Usually, this can be readily obtained from the indicator diagram, where it is shown by a distinct change or reversal in the curve, which may be more conclusively located by comparison with the true hyperbolic curve. But if not so obtainable, it should be ascertained at some convenient opportunity, from an inspection of the steam-valves of the engine. When the admission line and the expansion curve fade into each other without any defined point of separation, the weight of steam calcu-

lated from the point of cut-off is not very largely affected by a moderate error in location of the point upon the diagram, but it should be determined with certainty, if possible.

Expansion of high-pressure steam.—Steam of high pressure can only be employed to advantage by the adoption of expansive working. For reasons elsewhere treated, it is not found practically desirable to cut off steam in any cylinder before 30 to 35 per cent. of the length of stroke has been accomplished. Consequently the same steam is applied in several cylinders in succession, and the most suitable proportions for these are obtained by calculation. The number of cylinders or stages in which the steam is appropriately utilized depends upon its pressure. Approximately, one stage for each 50 or 60 pounds pressure may be adopted. The drop from the terminal pressure to the back pressure in any cylinder should be from 3 to 5 pounds. If the drop is greater, there is a loss of work which might be obtained from the steam. If, on the other hand, the drop is less, the amount of nett effective pressure obtained at the end of the stroke is insufficient to drive the engine, so that for the moment the engine consumes instead of produces work, by reason of its motion, and rather more useful work would be produced by the adoption of a reduced stroke, while retaining the same diameter of cylinder, and using the same amount of steam. As the pressure is increased and more stages of expansion adopted, the difference or range of temperature in each cylinder becomes reduced.

Estimation of pressure in receiver.—In the calculations for a compound or a triple or quadruple expansion engine, the pressure in each receiver is required. Unlike the pressure in the steam-chest supplying the first cylinder, the pressure in each receiver varies

considerably. In ordinary cases, in which the cylinder supplies steam to a receiver, and one other cylinder is supplied by the receiver, the pressure in the receiver rises after the steam is cut off in the cylinder which takes steam from the receiver. This increase in pressure is due to a true compression which occurs in the first cylinder and the receiver, which, during the period in question, are generally in free communication. The receiver pressure therefore rises from the moment of cut-off, until the next succeeding point of admission is reached; and, conversely, the receiver pressure falls during the period in which steam is supplied from the receiver to the later cylinder. Usually the back pressure in the former cylinder may be assumed to be equal to the pressure in the later cylinder, immediately after admission. During the period of admission to the later cylinder, the steam contained in it, that in the receiver and that in one end of the cylinder which exhausts into the receiver, expands together, truly and usefully. In many cases this expansion is merged in that which in the later cylinder succeeds cut-off. It then becomes difficult and sometimes impossible to distinguish between the two upon the diagram. In such cases the fall in pressure in the later cylinder immediately preceding cut-off is due to expansion much more largely than to wire-drawing or throttling at the valves. The amount of this fluctuation in pressure in the receiver may be calculated upon the combined capacity of the receiver and of the two cylinders in free communication with it at successive periods. In a triple expansion engine in which the three pistons are connected to separate cranks, set at angles of 120° with each other, the cranks passing the top of the stroke in the order—high, intermediate, and low pressure, in which steam is cut off in the later cylinder at 40 to 45 per cent. of the stroke,

and in which the capacity of the receiver is about twice as great as that of the cylinder which supplies it with steam, the difference in absolute pressure is about 100 : 75. When steam is admitted from a receiver to an intermediate or low-pressure cylinder in such a manner that, before cut-off, the pressure remains practically uniform, the true or useful expansion of the steam is suspended during such period, and the steam is simply pumped from one cylinder into another. The total amount of expansion is unaffected, but under such a condition it is irregularly distributed.

Weight of steam passing successive cylinders.—A certain amount of steam is supplied to the high-pressure cylinder in each stroke. This total amount of steam must pass through the successive cylinders without diminution or increase, except such as may arise from positive or negative leakage. But in its course through the engine the steam suffers partial condensation, so that in successive cylinders a reduced quantity of live steam passes per stroke. The weight of this steam in successive cylinders being known, or its proportionate reduction by condensation being assumed, and the cylinder capacity at cut-off being known, its pressure at such point is easily determined, and from this the pressure at admission and the back pressure in the preceding cylinder.

Steam-pressure in cylinder obtained by use of tables.—By the use of Table XV. only, the mean pressure is, as already stated, obtained, on the assumption that the pressure is quite uniform during the period of admission and up to the point of cut-off. But when the pressure falls appreciably during the period of admission, the direct use of the table involves some sacrifice in accuracy. If the factor from the table is multiplied by the pressure at cut-off, the result obtained is less

than the mean pressure throughout the stroke, while if multiplied by the pressure at admission, the result obtained is much more variable and unreliable. The quantity obtained from Table XV. may, however, be corrected by the addition of a quantity obtained from Table XVI. As an example, the mean pressure upon a piston may be assumed to be required, the initial pressure being 100 pounds per square inch, the pressure at cut-off 80 pounds, cut-off occurring at 45 per cent. of the stroke, and clearance amounting to 5 per cent. The factor from Table XV., corresponding to 45 per cent. cut-off and 5 per cent. clearance, is ·822. That from Table XVI., corresponding to 45 per cent. cut-off and a fall of 20 per cent. from the initial pressure, is ·075. The mean pressure throughout the stroke therefore =

(Pressure at cut-off = 80 lbs.) × (·822 + ·075 = ·897) = 71·76.

The quantities given in Table XVI. are generally approximate. They apply strictly to cases in which the area of the part of diagram shown by cross-hatching in Fig.

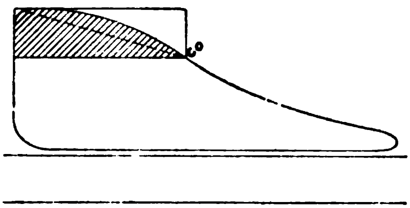

Fig. 29.—Diagram showing fall of pressure during admission.

29 is two-thirds that of the circumscribing rectangle. If the admission line is a perfectly straight line, as shown dotted in Fig. 29, any multiplier selected from Table XVI. should be reduced to three-fourths of its original amount before adoption.

TABLE XIV.—CYLINDER QUANTITIES.

1250 indicated horse-power required. Boiler pressure, 170 lbs. per sq. in., absolute. Steam expanded to 6·7 lbs. per sq. in. Back pressure, 3 lbs. per sq. in. Stroke, 5 feet. Speed, 70 revols. per minute.

a.		Cylinders. High pressure.	Cylinders. Intermediate.	Cylinders. Low pressure.	
	Pistons:				
b.	Diameter (in.)	21·00	33·00	55·00	Total area = 3506·23 sq. in. = 2·78 in. per horse-power per hour (nett).
c.	Nett area (piston and tail-rods 5½ in. diameter) (sq. in.)	322·60	831·55	2352·08	
d.	Vol. swept (cub. ft.)	11·20	28·87	81·67	
	Clearance:				
e.	Percentage ...	6·0	5·0	4·0	
f.	Cubic feet ...	·67	1·44	3·27	
g.	*Cut-off:* (per cent.)	38·0	44·8	50·0	
h.	*Dryness of steam:* (do.)	84·6	81·0	77·4	
	Live steam at cut-off:				
k.	Volume (cub. ft.)	4·93	14·37	44·10	
l.	Weight (lbs.)	1·678	1·607	1·536	1·988 lbs. water per stroke = 16,661 lbs. per hour = 13·23 lbs. per ind. h.-p. per hour.
	Absolute pressures:				
m.	Initial (lbs. p. sq. in.)	161·5	60·4	17·8	
n.	Cut-off (do.)	153·3	45·3	13·4	
o.	Terminal (do.)	63·6	20·8	6·7	
p.	Mean (do.)	119·6	41·7	13·0	
q.	Back (do.)	60·4	17·8	3·0	
r.	*Horse-power constants*	6·842	17·637	49·892	
	Indicated horse-power:				Total.
s.	Gross ...	405·0	421·5	498·9	1325·4
t.	Nett = $s \times$ ·95 ...	384·7	400·4	474·0	1259·1
	Temp. in cylinders:				
u.	Extremes ...	364° F. to 293°	293° F. to 222°	222° F. to 142°	
v.	Difference ...	71°	71°	80°	

TABLE XV.—AVERAGE PRESSURE UPON PISTON, CORRECTED FOR CLEARANCE.

Cut-off percentage	Clearance percentage upon volume swept by piston.					
		5.	10.	15.	20.	25.
a.	*b.*	*c.*	*d.*	*e.*	*f.*	*g.*
10	·330	·392	·441	·481	·515	·545
15	·435	·481	·520	·554	·582	·605
20	·522	·558	·590	·616	·639	·659
25	·600	·626	·650	·673	·692	·707
30	·662	·684	·704	·722	·737	·751
35	·717	·736	·751	·766	·779	·790
40	·766	·782	·794	·806	·815	·825
45	·809	·822	·831	·840	·849	·856
50	·847	·859	·864	·871	·876	·884
55	·879	·886	·892	·898	·903	·907
60	·906	·911	·916	·920	·924	·928
65	·930	·934	·935	·939	·942	·946
70	·950	·952	·955	·956	·958	·961
75	·966	·967	·969	·970	·972	
80	·978	·979	·980	·980		
85	·988	·988	·989	·989		
90	·995	·995	·996	·996		
95	·999	·999	·999	·999		
100	1·000	1·000	1·000	1·000	1·000	1·000

Calculations recorded in Table XIV.—The results of calculations may be recorded in Table XIV., which is completed for a triple expansion engine with three cylinders and three cranks, intended to give 1200 indicated horse-power at a consumption of 12·5 pounds of steam per indicated horse-power per hour, such steam being measured as water fed to the boiler. It is assumed that 84·6, 81·0, and 77·4 per cent. of the water supplied to the boiler will be found to exist as steam in the successive cylinders at points of cut-off, which figures

TABLE XVI.—ADDITIONS TO BE MADE TO AVERAGE PRESSURES IN TABLE XV., ON ACCOUNT OF FALL OF PRESSURE BEFORE CUT-OFF.

	Percentages of fall: $\frac{\text{Initial pressure}}{\text{Cut-off pressure}} = \frac{100}{100-\%}$						
	5.	10.	15.	20.	25.	30.	35.
a.	*b.*	*c.*	*d.*	*e.*	*f.*	*g.*	*h.*
10	·003	·008	·012	·017	·022	·028	·036
15	·005	·011	·018	·025	·033	·043	054
20	·007	·015	·024	·033	·044	·057	·072
25	·009	·018	·030	·042	·055	·072	·090
30	·010	·022	·036	·050	·066	·086	·108
35	·012	·026	·042	·058	·078	·100	·126
40	·014	·030	·048	·067	·088	·114	·144
45	·016	·034	·054	·075	·100	·128	·162
50	·018	·037	·060	·083	·110	·143	·180
55	·020	·040	·066	·096	·122	·157	·198
60	·021	·044	·071	·100	·133	·171	·215
65	·022	·048	·077	·108	·144	·186	·233
70	·024	·052	·083	·117	·156	·200	·251
75	·026	·056	·089	·125	·167	·214	·269
80	·028	·059	·095	·133	·178	·229	·287
85	·030	·063	·100	·146	·189	·243	·305
90	·032	·067	·106	·150	·200	·257	·323
95	·034	·070	·112	·158	·211	·272	·336
100	·035	·074	·118	·167	·222	·286	·359

are recorded in line *h*. The stroke of the engine is assumed to measure 5 feet, the speed to be 70 revolutions per minute, and the piston speed 700 feet per minute. Line *l* gives the corresponding weight of live steam in each cylinder per stroke, and at the end the weight of water supplied to the boiler = 1·983 × 70 × 2 × 60 = 16,661 pounds per hour. A terminal pressure in the low pressure is specified, which in this case is 6·7 pounds,

and a boiler pressure of 170 pounds per square inch, absolute. The volume of 1·536 pounds of steam, at a pressure of 6·7 pounds per square inch, furnishes the means for supplying the figures for the low-pressure cylinder in lines *b*, *c*, and *d*, due allowance being made on account of clearance. In the low-pressure and intermediate cylinders there is but little condensation and re-evaporation during expansion, consequently the weight of live steam remains nearly constant, and the points of cut-off may be selected so as to secure a suitable pressure at the time. In the example, the steam is assumed to be cut off at 50 per cent. in the low-pressure cylinder. The volume of steam at cut-off is then found to be 44·10 cubic feet, and inserted in line *k*. The pressure of 1·536 pounds of steam occupying this volume is 13·4 pounds per square inch. One-third added to this gives 17·8 as the initial pressure, and also the back-pressure in the intermediate cylinder. In the high-pressure cylinder the initial pressure may be assumed to be 5 per cent. below the full boiler pressure, and the pressure at cut-off 5 per cent. below the initial pressure. A suitable pressure at cut-off in the intermediate cylinder is a mean proportional between the pressures at cut-off in the high- and low-pressure cylinders, in this case $= \sqrt{153{\cdot}3 \times 13{\cdot}4} = 45{\cdot}3$, the initial pressure being again one-third greater, to give a fall of 25 per cent. before cut-off. The terminal pressure in the intermediate cylinder is obtained from the weight of steam and the total volume—including clearance—occupied by it. The terminal pressure in the high-pressure cylinder is affected by re-evaporation in the cylinder differently from those in the succeeding cylinders, and may be obtained with sufficient accuracy by proportion $= \dfrac{\text{Pressure at cut-off} \times \text{total volume at cut-off}}{\text{Total volume at end of stroke}}$

The terminal pressures in the intermediate and low-pressure cylinders may also be obtained in the same way without serious error. To secure best results, the terminal pressure in each cylinder should be 3 to 5 pounds above the back pressure. The mean pressures are then obtained as previously described. The back pressure upon the low-pressure piston is assumed to be 3 pounds. The constants in line r for the several pistons =

$$\frac{\text{Nett area of piston in sq. in.} \times \text{speed of piston in ft. per min.}}{33{,}000}.$$

They are equal to the calculated horse-power of each piston due to a mean effective pressure of one pound per square inch. In each case the gross horse-power is obtained from the mean effective pressure =

$$(\text{mean pressure} - \text{back pressure})$$

by multiplication with the constant. The nett indicated horse-power is obtained from the last by means of a suitable reduction on account of losses of pressure in ports and in other ways. In different cases this allowance may be from 5 to 15 per cent. or more, but in the present case it is taken at 5 per cent. in each cylinder. No allowance is here made on account of steam saved by the adoption of compression, nor for the indicated horse-power thus lost, as the one or the other of these may preponderate. The extreme temperatures in each cylinder are given at the foot of the table; also the differences, or the range of temperature in each cylinder.

Effects of variation in proportions.—In some trial calculations the initial pressure in one cylinder will be found to exceed the terminal pressure in the preceding cylinder. This may arise from the tentative adoption of an unsuitable size of cylinder in one case, or from

the adoption of too early cut-off in one or both cylinders, and corresponds to the case, occasionally met with, in which the back pressure exceeds the terminal pressure in the cylinder.

When in a triple engine working in a normal manner the steam is cut off at an earlier point in the high-pressure cylinder, the total amount of work done in the three cylinders is reduced almost in proportion to the reduction in the amount of steam used. The amount of work contributed by the high-pressure cylinder is slightly reduced. The work of the intermediate cylinder is more largely reduced. But the greatest reduction is in the work of the low-pressure cylinder. The consumption of steam per horse-power is appreciably reduced, by reason of the increase in the total expansion effected, but the engine is larger and more costly in proportion to the work done.

The work done by one of the later cylinders of a triple or quadruple expansion engine — or the low-pressure cylinder of a compound—is always increased by cutting off steam at an earlier period, and that of the preceding cylinder is almost equally reduced, on account of an increase in the receiver pressure. When steam is cut off earlier in the intermediate cylinder, the work of the low-pressure cylinder is little if at all affected.

When the cut-off in the low-pressure cylinder is accelerated, the amounts of work contributed by it and by the intermediate cylinder are affected just as described. The work of the high-pressure cylinder is unaffected, but, within moderate limits, the total amount of work done by the engine is usually slightly increased by the adoption of an earlier cut-off in the low-pressure cylinder.

Within moderate limits, each of the results set forth, as arising from the adoption of an earlier cut-off, is reversed by an opposite course. But in any case of

uncertainty, two or three trials in the form of Table XIV. will be useful.

An increase in the diameter and area of a cylinder without change in the point of cut-off causes a reduction in the pressure of steam corresponding to a certain weight of steam in the cylinder. This will cause a greater drop in pressure as compared with the terminal pressure of the next preceding cylinder. An increase in the area of a cylinder may be accompanied by such an acceleration in cut-off as to leave the pressure at cut-off quite unaffected, but the terminal pressure will in all cases be reduced, and may thus fall below the natural pressure of admission to the next following cylinder. The diameter of cylinders should be so adjusted, if possible, as to avoid fractional parts of inches. In comparing different engines, the total cylinder area is of interest, especially as compared with the work done. In the example given this works out to 2·78 square inches in nett area of piston per indicated horse-power. If the terminal pressure is not carried so low, a smaller low-pressure cylinder will suffice, the engine will be less costly, and probably a nett gain will be secured, but at some sacrifice in economy of steam.

Application of results of calculation.—The example is not given with the intention of showing the best possible proportions to adopt, but only to show how calculations may be made which will be of service, either in deciding upon the proportions and conditions to be adopted in a new engine, or in working an existing engine to the best advantage, either normally or when required for a period to work at increased or reduced power. Also to indicate the possible advantage to be secured by the reduction of cylinder condensation, whether by means of superheating, jacketing, or otherwise. For this purpose it will be necessary to know the percentage of dryness

of steam at the several stages, to be inserted in line *h*, from which changes in other figures will follow. The gross power obtained from a given weight of superheated steam will be greater than that obtained from saturated steam, which is subject to greater condensation in the engine, but such gain will be subject to a little reduction on account of more rapid fall of pressure in the cylinders during expansion. The amount of difference is, however, insufficient to cause the rejection of Tables XV. and XVI.

Distribution of power in several cylinders.—The turning of the engine will be somewhat better when the power applied to the several cranks is about the same in amount. As a rule, however, rather better economy is secured when the power developed in the intermediate cylinder is rather greater than that in the high-pressure cylinder, and that in the low-pressure greater still. The variation should, however, not exceed the ratio 8 : 9 : 10. Excessive condensation occurs in the cylinders of all engines which are too large for their work, especially in the low-pressure cylinders of compounds or triples. Such loss affects both the total economy and the equal distribution in the several cylinders.

Approximate character of prediction.—In calculations of importance it will probably be necessary to work through half-a-dozen or more separate sheets, applying to different sets of conditions. Owing to the large number of factors which enter into each case, and in every case more or less differently from all others, it is impossible to expect to predict with absolute certainty every detail as to the pressure at each point, and as to the whole of the conditions. But calculations made in the form of Table XIV. will be found to give all necessary information with sufficient correctness for practical purposes.

CHAPTER XXII.

PISTON SPEED AND LENGTH OF STROKE OF ENGINES.

Work done in proportion to speed of engine.—The amount of power produced by an engine of given dimensions, and supplied with steam of same pressure upon the piston, is correspondingly increased or diminished by a change in speed. In an ideal engine the consumption of steam would follow in strict proportion to the speed, but in practice several losses are incurred to modify such advantage. But by the adoption of a high speed, a smaller engine suffices for the performance of a given amount of work than would be necessary at a more moderate speed.

Effect of inertia due to weight of reciprocating parts at high speed.—Piston speed may be increased either by increase in speed of revolution or in length of stroke. Any increase is accompanied by a corresponding increase in the amount of force required to set in motion the whole of the reciprocating parts at the beginning of each stroke, and in that required to bring the same to rest at the end of each stroke. These parts include the piston, piston-rod, cross-head, and the connecting-rod for about two-thirds the length nearest to the cross-head. The total weight of these is considerable, which would appear to be objectionable on account of the power

absorbed. But this power is absorbed or stored in the inertia of the moving parts, and is given out again undiminished in the act of retardation of the motion during the latter part of the stroke, so that while on this account the pressure upon the crank-pin during the first part of the stroke is less than that due to the steam-pressure at the time acting upon the piston, the pressure at the latter part of the stroke is just as much greater than that due to the simultaneous steam-pressure upon the piston. In an engine working without expansion this gives rise to great irregularity in motion. In an engine which is worked expansively, the power of the steam is greatest at the beginning of the stroke, and least at the latter part. The force of inertia due to the alternate starting and stopping of the reciprocating parts, at a high speed, furnishes an excellent means of approximate compensation for the irregularity of steam-pressure. For this reason it is sometimes desirable to increase the weight of reciprocating parts beyond that due to the amount of material which is necessary on account of strength. In this way, a most satisfactory degree of uniformity of pressure upon the crank-pin and corresponding uniformity of motion are secured. For a treatment in detail of this question, reference may be made to Porter's work on the "Richards Indicator." This method of compensation would be most efficiently secured if the whole of the weight could be concentrated in the piston. But obviously this is impossible, and hence the necessity for ample strength, rigidity, and good fitting in all parts, so that the transmission of the stresses due to inertia may not cause any appreciable change in form of any detail, or the development of a slack fit, which will rapidly lead to noise and violent pounding in work. With perfect fit between the crank-pin and the connecting-rod brasses, no knock

or excessive wear occurs at any practicable speed. But wear takes place, though very slowly, and the least degree of slackness shows itself at once, and when once fairly started it increases rapidly, if not attended to. The danger arising from pounding is doubly destructive at high speeds, as more blows are administered, and each is applied with greater force than would be the case at more moderate speeds. The crank-pin also must maintain its true cylindrical form, or one part will take less than its fair share of pressure, and another will take more. The part exposed to greatest pressure will suffer momentary wear, which will be transferred to another part at a different period in the revolution. Thus all parts will be exposed to wear or will be attacked in detail, so that the ruin of the bearing proceeds rapidly. At high speeds there is therefore the most imperative necessity for strength and almost absolute rigidity in the crank-pin; and the same applies with almost equal force to the crank-shaft, and in a less degree to the cross-head pin. The valve-gear, pumps, and other details, are also exposed to similarly severe work. Consequently, in engines designed to work at high speeds of revolution, there is a necessity for workmanship, material, and design of the highest quality, and such engines are naturally more costly, in proportion to their size, than those which are only intended to work at a moderate speed.

High speed of piston may be obtained by means of a great length of stroke, in combination with a moderate speed of revolution. In this case the force of inertia, which may be so usefully directed in connection with an engine of high speed of revolution, is of less value. Strength and rigidity of detail are always valuable, but minor deficiencies in these respects are not so promptly followed by serious consequences as in engines working

at a high speed of revolution. Pounding is much more under control, wear is reduced, and the original cost of the engine and the cost of maintenance are less than for a high-speed engine of similar dimensions.

Value of a fly-wheel at different speeds.—The regulating power of a fly-wheel of given weight and dimensions varies according to the square of the number of revolutions, so that comparing two wheels of identical proportions, employed in connection with equal powers, but of which one is driven at double the speed of the other, it will be found that the more rapid wheel will effect a regulation of speed within limits only one quarter as great as those of the slow wheel. If, however, the more rapid wheel is used in connection with power in proportion to speed, or twice as great as the second wheel, the regulating power will be reduced to one-half of that in the first case, or only in strict proportion to speed. In either case, however, a smaller fly-wheel suffices for an engine of high speed than would be necessary for one of moderate speed of revolution.

Comparison of speeds of revolution of engine and shafting.—In ordinary cases the speed of revolution of the crank-shaft is less than that of the shafts which are driven from it; consequently, speed must be gained and power lost in the transmission. This is a cause of considerable loss in efficiency, which is more or less avoided by the adoption of high speed in the engine.

Large area of ports required at high speeds.—The area of cylinder-ports must in all engines be sufficiently large to avoid an excessive loss in pressure of steam passing through them on its way to and *from* the cylinder. Some amount of loss must always arise in this way, and the sum of the two losses is to be considered as affecting the amount of power developed by the engine. As a rule, this loss should not much exceed

one pound to one and a half per square inch in reduction of the steam-pressure or increase of the exhaust-pressure. The permissible velocities given in the chapter on valve motions correspond to a theoretical loss of pressure of about one-quarter of a pound per square inch, but this is always exceeded by a large amount, which varies in different cases according to the perfection of the design. In the best-arranged passages steam will lose pressure approximately as follows—

$\frac{1}{4}$	per cent.	of its absolute	pressure	at a velocity	of	60 ft.	per second.
1	,,	,,	,,	,,		120	,,
4	,,	,,	,,	,,		240	,,

The velocity of steam can only be kept within definite limits when the area of port bears a definite relation to the area of the cylinder to be supplied with steam, and to its working speed. In engines designed for high speed, the requisite area of ports causes the item of port clearance in the cylinder to assume great importance. This question is treated in another chapter, but it may be here repeated that the absolute amount of clearance depends chiefly upon the diameter and speed of piston, and that the loss by reason of clearance may be reduced by reducing the number of times which the clearance space is filled with steam. In the adoption of high piston speed this advantage may be secured by means of comparatively great length of stroke, which also gives an advantage in connection with cylinder condensation in ordinary cases, by reducing the amount of surface exposed in comparison to volume of steam during the period of admission, when cylinder surface condensation is most powerfully developed.

Good construction essential for high speeds.—An engine driven at a high speed is sure to declare any defects in construction which may exist. But evident defects ought not to exist in any engine, as even in a low-speed

engine they cause unnecessary loss, and wear and tear in working. Defective balance is one of the most obvious and inexcusable faults to which an engine is subject. Truth and inflexibility of bearings and sliding surfaces are of great importance, as also is area of surface sufficient to keep the pressure per unit of area within moderate limits. The presence or absence of these qualities may not in all cases be developed so as to strike an observer forcibly, but they affect the tendency to heating, and the life of the engine as a whole or in detail. Probably in no case is mischief more likely to become developed in secret than by reason of imperfect circularity in the bearings of shafts and pins. If other means are not available for ensuring correctness in this respect, grinding by an emery wheel should be resorted to. Vertical engines, of which the standards are constructed upon correct principles, are quite well adapted for working at a high speed. Horizontal engines are most popular. In these the same conditions cannot be so completely observed to secure the preservation of a true centre line of piston-rod, but they should nevertheless be kept in view. If the cylinder is arranged to overhang the frame, the possibility of bending of frame in front of the cylinder joint should be carefully considered. Depth of bed-plate is in all cases exceedingly important for the purpose of keeping the bending action within moderate limits. Overhanging bearings may not necessarily be objectionable, but such are placed at a great disadvantage in resisting vibration or irregular stresses arising from defective balance or condition of parts, or from irregular work, and should be avoided where possible.

The Allen engines, made by Sir Joseph Whitworth twenty-five years ago, work at a piston speed of 1000 feet per minute. They were beautifully made, and a

credit to all concerned in their production. Experience in their working gives support to the dictum of Mr. Porter that bearings and surfaces can be made which will not wear out. But the principle has met with only a moderate degree of acceptance. One reason for this is, no doubt, the original cost of such engines. Another is that only a limited number of engine builders are prepared to supply the quality of work which is requisite for success. Mr. Porter also recently at Chicago referred to the loss experienced by reason of excessive volume of ports and clearance, and to the risk of damage by water in the cylinder. Though the high speeds which were advocated twenty-five years ago have not been extensively realized in practice, the general soundness of the principle has been amply vindicated by the successful working of the engines made at that time. But their beneficial influence upon general practice is not measured by the extent to which they are absolutely copied. Most probably without them speeds would have advanced, but less rapidly and with less confidence on the part of owners and makers.

Variation of speed with length of stroke.—Having regard to the whole of the conditions, as to steadiness of running, moderate volume of clearance, character of workmanship generally available in construction, skill and attention necessary in attendance, original outlay, cost of maintenance, and working results, the adoption of the high speed of 1000 feet per minute should be generally avoided. The piston speed may be greater with longer strokes, not because of any difficulty arising from the reciprocation of the parts, but on account of the relative magnitude of the losses in the ports, owing to the smaller number of times they require to be filled. With good design and construction, speeds in accordance with Table XVII. may be adopted with confidence.

TABLE XVII.—ADVANTAGEOUS SPEEDS OF ENGINES.

Stroke, 6 in.	Revolutions per min., 450.	Piston speed, 450 ft.
do. 1 ft.	do. 240.	do. 480 „
do. 2 „	do. 140.	do. 560 „
do. 4 „	do. 80.	do. 640 „
do. 6 „	do. 60.	do. 720 „
do. 10 „	do. 42.	do. 840 „

These speeds may be increased 20 per cent. for engines of exceptionally good construction. On the other hand, they may be reduced 20 per cent. without serious loss. Still lower speeds may appear to present points of convenience, but are better avoided.

CHAPTER XXIII.

VALVE ARRANGEMENTS FOR STEAM-ENGINES.

Slide-valves for admission of fluid to cylinder and withdrawal of same.—Steam or other fluid, supplied under pressure, is applied for the purpose of producing motion. With this object, it is admitted alternately to one or both ends of a cylinder in which a piston is fitted, and to which motion is imparted by the fluid. The admission of the fluid to perform work, and its subsequent release from the cylinder, are controlled by

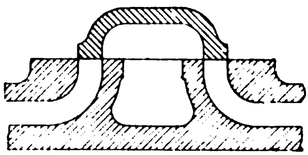

Fig. 30.—Slide-valve without lap, in central position.

valves, of which the common or locomotive slide-valve is probably the most largely adopted. Such a valve sustains the full supply pressure on the outer or upper side, while the inner or lower side is always exposed to the back pressure. The pressure gives rise to a large amount of friction, and consequently a valve of this type requires a comparatively large amount of power to drive it. A rudimentary valve of this type is shown

in Fig. 30, in its central or closed position. When the valve is moved to the left, as in Fig. 31, the port *a* is placed in communication with the supply, and simultaneously port *b* is placed in communication with the cavity *c* of the slide-valve, and with *d*, the exhaust-port. In this way the motive fluid will be allowed to flow into one end of the cylinder to perform work, while simultaneously the spent fluid will be allowed to escape from the opposite end of the cylinder, by means of the exhaust-port. In all positions of the valve except the central one the same condition exists, one port being open to supply and the other to exhaust. Subject to slight corrections on account of angularity of connecting-rod and eccentric-rod, such a valve must

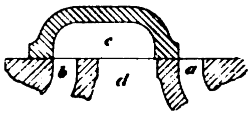

Fig. 31.—Slide-valve without lap, in extreme position.

always occupy its central position when the piston is at either end of the stroke, and the extreme position when the piston is in its central position. Usually, but not necessarily, the amount of travel, or the distance by which the valve is moved, is such that in its extreme position the port is just completely open, as in Fig. 31, the opposite port being similarly opened at the opposite end of travel. Such a valve is suitable for use under water-pressure, or with fluid of low elasticity, no provision being made for expansion.

Expansion obtained by slide-valve.—If the faces of the valve shown in Fig. 30 are extended, as in Fig. 32, expansion becomes possible, and the valve is suitable for use with steam. A considerable period exists during which both supply-ports are closed; the valve

must therefore move a definite distance from the central position before either port is opened. The mechanism which actuates the valve is so adjusted as to open the port nearly at the beginning of the stroke, as in Fig. 33, so that the whole of the reduction of the time during which the port is open applies to the last part of the

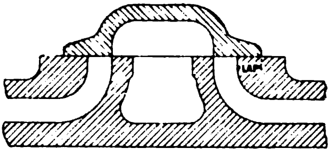

Fig. 32.—Slide-valve with lap equal to width of port, in central position.

stroke, when the cylinder is full of steam ready for expansion. The additional width of valve, or the distance by which the valve is required to move from its central position before commencing to open the port is called the "lap" or "outside lap" of the valve, as shown in Fig. 32, and is expressed in lineal measure. It is found to be necessary for the valve to open the port, a little before the exact moment at which the piston commences its stroke, by which means the engine is found to work more steadily and silently.

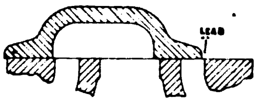

Fig. 33.—Slide-valve with lap equal to width of port, showing lead.

The amount of this opening is called the "lead" of the valve, and is also expressed in lineal measure, as in Fig. 33. An increase in the amount of lap causes an increase in the length of time during which the supply of steam is shut off from the cylinder, and therefore an increase in the degree of expansion. But

unless the amount of travel of the valve is increased simultaneously with the lap, the opening of the port will be reduced. In many cases one of the conditions demanded is that the valve shall open a full port to the steam. But in other cases a slight sacrifice is made in this respect, so long as the full area of port is main-

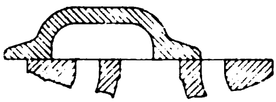

Fig. 34.—Slide-valve with lap equal to width of port, in extreme position, showing full port opening.

tained to exhaust, at which time the steam is of lower pressure and greater volume than at the time of its entry into the cylinder. In Fig. 32 the amount of lap is equal to the width of port. When in such a case the amount of travel is just sufficient to open a full port, as in Fig. 34, the steam will be cut off at about 73 per cent. of the stroke, subject to a slight correction on account of lead. When with such a valve this degree of expansion is exceeded, the full opening of the port is generally not accomplished. Applying this to the same

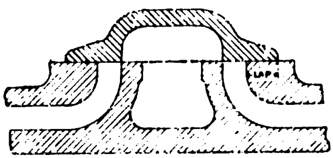

Fig. 35.—Slide-valve with lap equal to $1\frac{1}{3} \times$ width of port, in central position.

ports as in Fig. 32, giving the same travel to the valve, and causing a port opening of two-thirds the width of port, the cut-off will be effected at about 54 per cent. of the stroke. Fig. 35 shows such a valve in its central position, and Fig. 36 in its extreme position. Beyond this point it is not desirable to increase the expansion by

means of a plain slide-valve, on account of the excessive amount of friction which is encountered, by reason of the length of travel; and also on account of the loss of pressure by wire-drawing the steam before cut-off. Usually, the dimensions of the exhaust cavity of the slide-valve are not affected by the application of outside lap; and the valve remains of such proportions that in all positions except the central one it connects one of the cylinder-ports with the exhaust-port. Each port is therefore open to exhaust for half of the total time as before, but distribution is changed. In the case of the rudimentary valve without lap shown in Fig. 30, the exhaust is opened simultaneously with the steam supply, at the end of the stroke, at which time the

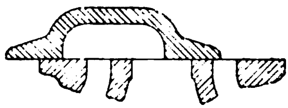

Fig. 36.—Slide-valve with lap equal to $1\frac{1}{3}$ × width of port, in extreme position, showing $\frac{2}{3}$ port opening.

valve is at the centre of its travel. When lap is applied, the eccentric is moved forward, or in "advance," so that the steam may be admitted at the same moment as it would in the absence of outside lap. By this means also the exhaust is advanced, so that the steam is released before the end of the stroke is reached, as is shown in Fig. 33. The proportion of length of stroke performed before the exhaust is opened can be easily ascertained by geometrical construction, and is such that the *time* by which the release is advanced is exactly the same as that by which the opening of full port to steam supply is advanced. The steam being released before the end of the stroke, and each cylinder-port remaining open to exhaust for half

the time of one complete revolution, it follows that the port is closed before the end of the return stroke, and any steam then present is retained and compressed by the advancing piston until the stroke is completed. The amount of this compression in the cylinder may be increased by reducing the width of the exhaust cavity giving "inside lap," as in Fig. 37. When the compression is increased by the adoption of inside lap, the point of release is postponed. Opposite results are obtained by increasing the width of the exhaust cavity or giving "negative inside lap," but this is seldom adopted except in locomotives or other engines running at a very high speed, as a measure for reducing the back pressure.

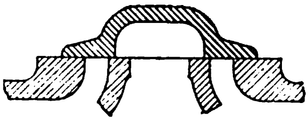

Fig. 37.—Slide-valve, showing inside lap.

Divided slide-valve.—If the slide-valve of the type just described is provided in connection with a cylinder of great length of stroke, the cylinder-ports, reaching nearly half the length of the cylinder, will be difficult to make clear and smooth; and when made they will give an excessive amount of clearance to be filled with steam at each stroke, which condition is opposed to economical working. To avoid these objections, the cylinder-ports are shortened, as in Fig. 38, the exhaust-port is duplicated, and the slide-valve is divided into two portions, each of which is practically equal to one-half of the original, with the addition of a plain end. Two valves of this type applied to a cylinder present greater surface to pressure than one of the former type; and consequently a greater amount of frictional resist-

ance is encountered. The comparatively cool exhaust chamber also acts upon the cylinder—and the steam within it—as a refrigerator. But the advantages secured by a reduction of port resistance and of clearance are greater than the disadvantages. This type of valve has

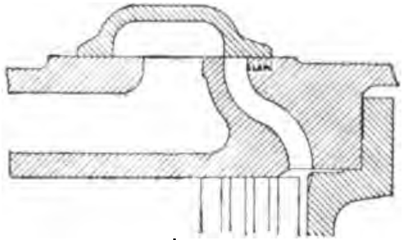

Fig. 38.—Divided slide-valve, in central position.

therefore been very largely adopted for cylinders of comparatively great length, in which the degree of expansion adopted is not too great for the application of plain slide-valves. Figs. 39 and 40 show the extreme positions of such a valve when in work.

Velocity of steam through ports.—The velocity of steam passing through the ports requires to be limited, with a view to avoid loss of pressure, and to avoid the

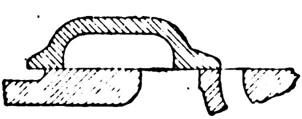

Fig. 39.—Divided slide-valve, in extreme position, showing full port opening.

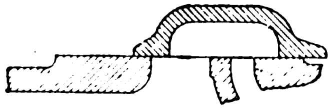

Fig. 40.—Divided slide-valve, in opposite extreme position.

mechanical cutting of the surfaces which sometimes occurs. In a very well-designed arrangement of ports, through which steam is passed at a velocity of 120 feet per second, a loss of 1 per cent. in the absolute pressure will be incurred. If the velocity is allowed to

reach 170 feet per second, the loss will be 2 per cent.; and at a velocity of 208 feet the loss will be 3 per cent. This is approximately the loss which occurs in the port, but the steam loses pressure at each bend which it encounters, also by reason of contact with rough surfaces, and by passage through enlarged and contracted channels. Having regard only to the loss arising in the ports, it is found that the following velocities should not be exceeded—

Steam of absolute pressure	200 lbs.	per sq. in.	should not exceed	80ft.	per second.
,,	100	,,	,,	100	,,
,,	50	,,	,,	125	,,
,,	25	,,	,,	160	,,
,,	12½	,,	,,	200	,,
,,	6	,,	,,	250	,,
,,	2½	,,	,,	340	,,

At the lower pressures some advantage would be secured by the adoption of lower velocities than those given, but at the cost of providing and working larger and more cumbrous valves, and incurring greater losses by reason of clearance. The area of exhaust-ports must be calculated upon the volume of steam when expanded to the pressure of exhaust.

Variation in velocity at different periods in stroke.—In all cases, the velocity of steam through the ports varies at different parts of the stroke, but especially is this the case when the valves are of the ordinary slide pattern, driven by eccentric. The length of *time*, from the moment at which the valve commences to open the port to the moment at which the closing of the port is completed, may be divided into two equal parts. During the first period the opening of the port is continuously increasing, while during the second the port is continuously closing. The first period partially coincides with the first half of the stroke of the piston.

The velocity of the piston is at first very low, but rapidly increases as the stroke advances, causing the demand for steam measured per unit of time to correspondingly increase. The increasing area of port opening very closely coincides with the increasing demand for steam, until the maximum opening of port is reached, and consequently during this period the variation in the velocity of steam is unimportant. With a valve as shown in Fig. 30, without lap, and consequently admitting steam throughout the stroke, the maximum opening would be reached at half-stroke, after which the valve would close precisely as it opened, and the velocity of steam throughout the stroke would be practically uniform. In ordinary practice, the maximum opening of steam-port is reached before the piston arrives at the centre of its stroke, after which, though the demand for steam to fill the cylinder continues to increase slightly, the port opening decreases. When steam is cut off at half-stroke, the velocity of the piston and the consequent demand for steam increase quite up to the time of cut-off, though for the latter part the increase is slow. But the port opening only increases during half of such time, after which it diminishes so rapidly that only a partial supply of steam is possible, and the pressure falls. A similar action, rather less pronounced, takes place in all cases in which an ordinary slide-valve is used to cut off at a later period in the stroke. An example may be taken in which a cylinder of 18 inches bore, 6 feet stroke, working at the slow speed of 40 revolutions per minute, is provided with a steam-port $12 \times 1\frac{1}{2}$ inches, and a valve with lap equal to the width of the port. The length of time during which the port is open may be divided into eight equal parts, and assuming that the full steam-pressure is maintained upon the piston, it is found that the average

velocities during the successive periods are as follows—108, 116, 128, 144, 168, 210, 308, 997 feet per second. The required velocities are so high in the latter part of the time as to entirely prevent the continuance of the full pressure in the cylinder; hence the fall which is always revealed by the indicator as the point of cut-off is approached. The action of a plain slide-valve is thus seen to be quite efficient in providing sufficient area for the admission of steam during the early part of the stroke, but to be unsatisfactory in this respect during the short time preceding cut-off.

Cut-off slide-valves.—In many cases plain slide-valves are used to regulate the admission of steam, while the

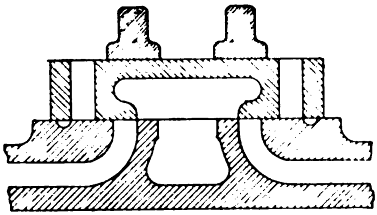

Fig. 41.—Cut-off slide-valve.

cut-off is effected by a second valve, generally working upon the back of the main valve, and in conjunction with it. One form largely used is shown in Fig. 41, in which the main valve is suitable for the same ports as are shown in Fig. 30, but the bearing surface is extended slightly. The cut-off valve is divided, each portion being adjusted by a screw, by which the distance apart is regulated, and consequently the time at which cut-off occurs. The cut-off valve is actuated by a different eccentric from the main valve. As a rule, the cut-off is effected at a time when the cut-off valve is moving at a high velocity, and the main valve is either stationary or is moving in the opposite direction.

Thus the process of cutting off is effected with rapidity, full area of port is maintained to a late period, and the loss of pressure is avoided or very largely reduced. The cut-off valve must be sufficiently wide to prevent admission of steam becoming re-established at the inner edge before the cylinder-port is closed by the main valve. In some modifications of the above, the cut-off valve is carried on the back of the main valve, and moves with it, until its motion is arrested by contact with stops acting in combination with levers, by which means the cut-off valve is rapidly thrown forward, and the cut-off effected. In Gonzenbach's modification, a separate stationary plate is provided, with ports cut

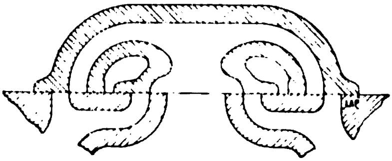

Fig. 42.—Double-ported slide-valve, in central position.

through, upon which the cut-off valve slides. The steam separated by the cut-off valve fills the inner section of the valve-chest, and consequently a considerable loss by clearance is incurred in the use of this arrangement, though the steam left in the chest at the end of the stroke is not allowed to entirely escape to exhaust, but is retained by the closing of the main slide-valve at a later period.

Double-ported slide-valves.—A modified form of slide-valve is made, by which the steam is admitted at two, three, or four points simultaneously, by which means a moderate degree of expansion may be secured, without an excessive length of travel. The common double-ported valve, as shown in Fig. 42, is largely used

in marine engines, chiefly for the intermediate cylinder of triple expansion engines, and for low-pressure cylinders. The same valve is shown in an extreme position in Fig. 43, giving full opening to steam. The Trick-valve, shown in Fig. 44, is used for the same reason on the Continent, chiefly for locomotives and other quick-

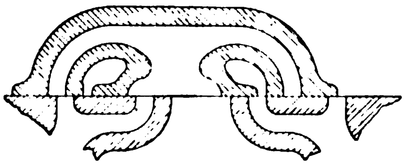

Fig. 43.—Double-ported slide-valve, in extreme position.

speed engines. It is adapted for higher expansion than the last. The proportions adopted should be such as to avoid possibility of transfer of steam from one end of the cylinder to the other by the valve passage when the valve is in its central position. In connection with this valve the exhaust is undivided, and therefore more free than in connection with the last valve.

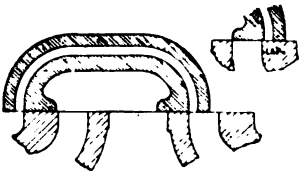

Fig. 44.—Trick-valve, in extreme and central positions.

Relations of lap, travel, and degree of cut-off.—The proportion of stroke at which cut-off occurs may be obtained in precisely the same way for all valves shown in Figs. 32, 35, 38, 42, 44, 45, and 46. The operation may be performed either by calculation or by graphic construction. Of the latter, Zeuner's method is the

best. Table XVIII. will, however, meet all practical requirements. The quantities required are the lap as defined upon Figs. 32, 35, 38, 42, 44, 45, and 46, and the length of travel of the eccentric-rod. The former of these quantities is to be divided by the latter giving a fraction which may be found in the first column of Table XVIII., or interpolated between two others. The proportionate point in the stroke at which cut-off occurs will then be found in the second column, subject to small corrections on account of lead, and angularity of connecting-rod and eccentric-rod. If the symmetry of motion of the eccentric-rod is interfered with by the use of bell-crank levers the table *may* not apply, and it certainly will not apply if a wrist-plate or its equivalent be adopted.

TABLE XVIII.—CUT-OFF BY SLIDE-VALVE.

$\frac{\text{Lap.}}{\text{Travel.}}$	Percentage of cut-off.
·00	No expansion.
·05	97
·10	93
·15	89
·20	82
·25	73
·30	62
·35	49
·40	35
·45	18
·50	No admission.

Friction of slide-valve.—Slide-valves of all kinds are subject to great resistance in work by reason of the friction which occurs at the faces. The amount of this cannot be accurately estimated, but the gear for driving such valves should be sufficiently powerful to overcome

the maximum resistance which can arise, and which is due to the full steam supply pressure on the back of the valve relieved by reason of minimum exhaust-pressure upon the face. This acts effectively upon rather less than the full area over cavity and bearing surface of valve face, but such relief is too uncertain to rely upon, and in any case is small at certain parts of the travel. The co-efficient of friction in this case is about ·15. The maximum resistance in pounds therefore =

$$\text{Total area of valve in inches} \times (\text{steam-pressure} - \text{exhaust-pressure}) \times \cdot 15.$$

The loss of horse-power incurred in moving the valve is liable to reach—

$$\frac{\text{Resistance in pounds} \times \text{travel in inches} \times 2 \times \text{number of revolutions per minute.}}{8 \times 33{,}000}$$

In large valves especially, as shown in Fig. 38 (of which two are required for each cylinder), this loss of power becomes very important.

Slide-valves balanced for pressure.—Many designs have been made with a view to reduce the resistance upon the slide-valve, by means of balancing the pressure, with some degree of approximation. This is chiefly effected by providing a closely-fitting surface at the back, of suitable area to relieve the pressure upon the face; but not on any account, or for one moment, to lift the valve out of actual contact with the face. The arrangement must give fairly uniform relief; if applied with greater force to alternate ends of the valve it may cause irregular wear of the surfaces, and possibly great leakage past the valve. It must be little affected by wear of the faces, and must provide sufficient elasticity so as to avoid the production of an additional element of friction by reason of excessive tightness, either constantly or at a particular part of the travel. Some arrangements provide very efficient relief to the valve,

but introduce almost, or quite as much, resistance in the relief-gear, so that no ultimate benefit is secured. Probably the most efficient relief-gears for this purpose are those in which a steam-tight piston is provided opposite to the slide-valve, and connected to the latter by means of a vibrating link. The space behind the relieving piston is connected with the exhaust-port by means of a pipe, which is provided with a cock for testing the steam-tightness by observations upon the gear when the cock is shut. Relief-frames are also made, in which the necessary elasticity is obtained by means of flexible steel diaphragms. When well designed these are efficient, but they are subject to great changes in form when under steam-pressure, and hence are difficult to adjust when free from pressure, being then either much tighter or much slacker than when at work. These should also be provided with a pipe and cock connected with the exhaust-port. A leak-hole may be drilled in the back of the valve to serve the same purpose, but this is not accessible for the purpose of testing.

Distribution of wear of surfaces.—The perfect action of a relief-frame or equivalent device, and indeed of a slide-valve generally, depends very largely upon the accuracy of the surface, not only in their original condition, but after a period of work when some loss of substance by wear has taken place. An important element in this connection to be regarded is that each pair of surfaces in contact shall, where possible, work so that each surface alternately projects beyond the other. The most perfect arrangement of sliding surfaces is one in which two uniformly loaded straight surfaces are in contact, in which they are of equal length, and exactly cover each other when at half-stroke. Such surfaces are arranged in the most favourable condition for preserving their rectilinear character under heavy

wear. Though slide-valves cannot usually observe such conditions, they may often be improved in design by paying regard to the ideal conditions of sliding surfaces. When a narrow surface must work against a broad one, and the limits of motion are sufficiently accurately defined, a good result is secured if the narrow surface reaches about half its width beyond the wide one, at each end of its stroke. In old engines, the slide-valve surfaces are sometimes found to be very much at variance with these conditions, and to consequently wear out of truth to such an extent as to cause enormous loss of steam. In many such cases it is found that a part of the surface of a plain slide-valve, about half-

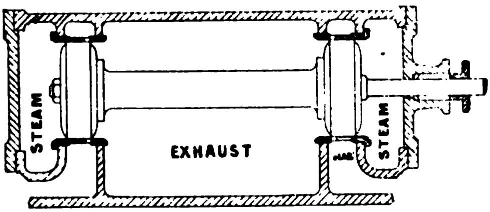

Fig. 45.—Piston-valve without central passage, in central position.

way between the exhaust cavity and the edge of the lap, is exposed to very little wear, by reason of the deficient amount of travel, either originally given or as arising by wear of parts. The remainder of the valve is worn more rapidly, so that truth of surface is lost, and the corresponding surface against the cylinder also suffers.

Piston-valves.—In piston-valves, the system of balancing is most perfectly developed. Two pistons are attached to one stem at a suitable distance apart, and arranged to work in bored chambers, as in Fig. 45. The inner parts of these chambers are exposed to exhaust-pressure, and the outer parts to steam-pressure.

Ports are provided in each chamber, and each piston is moved so as to place its cylinder-port alternately in communication with the steam supply and the exhaust. The two pistons together effect a steam distribution precisely as one short plain slide-valve or a pair of divided valves. The pistons are kept tight by means of packing-rings, corresponding to those in a working piston. Ramsbottom rings and other patterns of packing are used, also solid pistons with grooves cut around the edges. Probably one of the most satisfactory methods, where space admits of its application, is a solid ring made in halves, bolted together by means of inside lugs, between which a thickness of paper is inserted when the rings or the bores of the chambers become worn. Diagonal bars are placed across the ports to prevent the rings from spreading, and fouling the edges of the ports. Each ring must cover a greater width than the width of port, to prevent the leakage of steam past the valve. The solid casting of the valve-chest may be bored to suit the diameter of the piston-valves, but a much better arrangement is to provide separate chambers, well fitted and tightly pressed into position. By this means a more hard, dense, uniform, and durable surface is secured, the bars across the ports are more conveniently and perfectly provided, if necessary, the whole can be easily renewed, and generally the conditions of the design are much less restricted than when the whole must be cast together. In other cases, and especially in valves of large size, the two pistons are made in one casting, connected by a tubular portion, through which the steam passes from one end of the valve-chest to the other, as the most convenient means of steam supply, and dispensing with steam-pipe connection separately to each end, which is necessary in the former case.

Fig. 46 shows such a valve in its central position, and Fig. 47 in one extreme position, showing the fullest opening for steam. The packing-rings are required precisely the same in either case. In the majority of cases the small diameter of the piston and the great strength of ring required preclude any

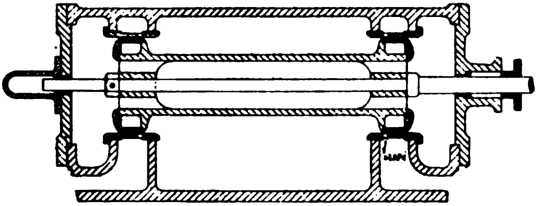

Fig. 46.—Piston-valve with central passage, in central position.

possibility of passing the ring over the solid piston, and junk-rings are required. These and the solid flanges at the opposite side must be kept clear of the ports, and are therefore coned down, as shown in the figures. The superior condition of balance which is possessed by piston-valves renders practicable a higher degree of expansion than is generally adopted with

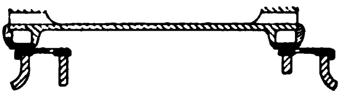

Fig. 47.—Piston-valve with central passage, in extreme position.

plain slide-valves. Table XVIII. also gives the degree of cut-off obtained with piston-valves when worked by eccentrics in the usual way. Piston-valves are generally designed to give an unnecessary amount of clearance, and consequently fail to give the best economical results. They are also rather troublesome to keep in order. For these reasons they are, as a rule, only

adopted for the high-pressure cylinders, but their use is extending, especially in vertical engines, in some cases for all cylinders.

Actuation of valves.—The whole of the valves hitherto described are usually actuated by means of eccentrics. In stationary engines each eccentric usually drives one or more valves directly. But marine and locomotive engines, and occasionally stationary engines, are required to revolve with equal facility and efficiency in either direction. For this purpose the device most frequently adopted is the provision of two eccentrics for each valve, the eccentric-rods of which are attached to a link, which allows either eccentric-rod to be placed into position for direct actuation of the valve. Other arrangements are also adopted, which are very simple in themselves, but in the discussion of which complicated geometrical problems arise. As a rule, the same arrangement which provides the means for reversing the direction of motion also provides for expansion of steam at lower powers.

Valves balanced for weight.—The slide-valves of all large engines require to be very rigid in every direction, and consequently are of great weight. In horizontal engines, the mere weight leads to considerable wear of the lower edge and of the facing upon which it works in the valve-chest. In vertical engines, the weight of the valve when rising increases the load on the valve-gear primarily due to friction; but relieves it when falling. This weight is often successfully balanced by means of a piston fitted to the top end of the valve-spindle, and working in a cylindrical chamber above the valve-chest. The balance-piston is provided with some ordinary kind of packing-rings, and the space above it is connected by a pipe to the condenser. This measure is precisely equivalent to a weight balance, with the exception that the tension applied by a rising

weight is different from that imposed by a falling weight, while the steam-pressure is uniform under all conditions. All valves may be balanced for weight in this way, whether balanced for pressure or not. Perhaps it is more desirable where the valves are balanced for pressure, as in such cases the valve motion is apt to be of lighter construction than would otherwise be the case.

Valves checked or actuated by a separate piston.—In some cases a cylinder is provided, very similar to that adopted for weight balancing, but closed at each end, so that the contained air or steam becomes compressed at each end of the travel of the valve, and the inertia of the mass is absorbed, whereby the valve is relieved from strain. This measure is especially useful in engines working at a high speed. In other cases, steam is admitted to such a cylinder in a manner to assist motion. In some engines a small cylinder is provided, to have sole control of the main valve, the small cylinder being supplied by steam under control of an eccentric or other motion directly derived from the crank-shaft.

Double-beat valves.—Double-beat, equilibrium, or "stamper" valves have been used very largely in Cornish practice, and in American river steamers; also to some considerable extent in stationary rotative engines. These possess some points of similarity to piston-valves, the chief difference being that they close against seatings, instead of providing an edge fit against the bored surface of a cylindrical chamber. If only one disc were employed, with steam-pressure upon its upper side, it could only be opened in opposition to the full pressure, acting upon its total surface. The application of pressure in opposite directions upon two discs gives perfect balance, subject to correction only on account of

the slight difference in area which is necessary for convenience in lifting out the valve and replacing it without disturbance of seatings.

Application of double-beat valves.—Double-beat valves are often driven by eccentrics, through a weigh-shaft, on which cams or wipers are mounted, which may be arranged to lift each valve with moderate rapidity, maintain a full opening for a sufficient length of time, and then allow the valve to fall with great rapidity for cutting off. In Cornish practice the valves are actuated by pegs on a plug-rod, which is driven by the beam of the engine. The constant head or resistance against which such engines work is such that the steam-pressure, in the absence of a good vacuum, is unable to effect motion. The vacuum depends upon the supply of injection water, which again is controlled by an adjustable cataract. It is obvious that an engine of this type must work at a very slow speed.

Many stationary engines were constructed from twenty to forty years ago, by first-class makers, in which double-beat valves were adopted. These were mostly beam-engines, and the valves were actuated by cams mounted upon a rotating shaft driven by toothed wheels, the cams giving every facility for adjusting the drop of the valve for cut-off. While in good condition throughout, this arrangement may be almost described as perfect; but it has hitherto given great trouble by reason of leakage past the valves. Unless the workmanship upon the valves, seatings, and preparation for seatings, is of the highest class, the attainment of a fair result is quite hopeless. The valves and seatings should be made from the same mixing of the same metal, so that no difference shall arise in the expansion and contraction by temperature. The upper and lower seatings of the same valve must be cast and fitted in one solid piece.

The thickness of metal, especially in the seatings, must be great, in order to impart rigidity and strength; it should also be as uniform as possible, to prevent warping by internal strain. Proell's valves are shown in *Engineering*, vol. xli., p. 250, and Sulzer's valves in vol. xlviii., p. 428. Prof. Riedler's valve is shown on p. 217, Pt. 2, *Elements of Machine Design*, by Prof. Unwin. In each case it will be seen that very heavy seatings are adopted. The assembling of the parts also calls for the greatest care. If putty or other luting substance is used in the joints, and any irregularity in its thickness or consistency should arise, the valve affected is almost sure to prove leaky. The joints should be so arranged as to avoid the necessity for the application of great force to make tight. But the best arrangements are open to abuse, and the least degree of carelessness or want of skill displayed in taking apart for examination, or in afterwards replacing the parts, may lead to unsuspected leakage. It is unquestionable that English experience with double-beat valves has hitherto proved unfortunate. Messrs. Sulzer of Winterthur, and other continental engineers, have, however, adopted them largely. When double-beat valves are used, separate valves are necessary for admission and exhaust. As a rule, this leads to a rather large percentage of clearance, but some of the engines made on the Continent and provided with such valves have proved to be remarkably successful and economical in work.

Corliss valves.—Corliss valves were introduced into England about thirty years ago, since which time they have made steady progress in favour, so that at the present time they are used more largely than any other upon stationary engines of high class and large sizes. In this system four separate valves are used, one for steam and one for exhaust, at each end of each cylinder.

These are placed in bored chambers, and are actuated by revolving about the centre. Each valve is cut away so as to allow the steam to flow past it, when it is turned into a position to open the port. They are free from excessive friction and wear, and are easily worked and maintained in good condition. As a result of their rotative actuation, the driving-rods may be attached to levers, of a length much greater than the radius of the valve surface. The valve-rods are therefore exposed to moderate stress, and may be made of light proportions.

Trip motion for Corliss admission-valves.—The steam-valves are opened by a direct pull, which overcomes the frictional resistance and also the opposition presented by a strong spring or a dash-pot. At a certain point in the stroke, generally controlled by the governor, the valve is detached from its positive connection, when the closing is effected by the spring pressure. A dash-pot is provided to steady the valve during closing by means of fluid pressure, sometimes regulated by the partial escape of the fluid through an orifice, or one dash-pot may replace the spring and also steady the valve during closing. The dash-pots are usually filled with air, which may be alternately rarefied and compressed; or the air may be exhausted to leave a vacuum; or oil may be used for slow work. The first Corliss engines made in England were fitted with wrist-plates, driven by one eccentric to each engine. The wrist-plate *may* transmit the motion unchanged in character from the eccentric to the valve. In such a case, the valves will be opened with an increasing velocity of motion until half of the travel has been covered, when the velocity will become reduced exactly as it was increased. The return stroke will be made precisely in the reverse order. But the wrist-plate provides the means whereby the motion imparted to the valves may be very largely

modified, and this object is usually the one intended to be secured by its use. The leading feature of the whole is the facility for giving a pause to the valve when fully open, so as to facilitate the passage of steam to and from the cylinder. By this means also the operations of opening and closing the ports are more rapidly and efficiently performed, and small ports become equally efficient with larger ports under different conditions. The wrist-plate is usually arranged with four studs to control the four valves of one cylinder. But two wrist-plates driven by separate eccentrics are often adopted with advantage, one for the steam- and the other for the exhaust-valves. The wrist-plate may be replaced by bell-crank levers with precisely the same result, as in the Corliss engine at the Centennial Exhibition of 1876. In the majority of recent engines, however, the wrist-plate or its equivalent is omitted, and the same ends are secured by other means, chiefly by giving a little more travel to the valve, in which the comparative freedom from frictional resistance in such valves is of great assistance. But whether or not a wrist-plate is used, steam-valves are almost always fitted with detaching gear to allow rapid closing, and to prevent wire-drawing—or reduction of pressure in the steam—during the period immediately preceding the closing of the valve. The variations in design which affect this section of a steam-engine are perhaps more numerous than in any other. About one hundred and forty types, chiefly taken from the Paris Exhibition of 1878, are described in *The Corliss Engine and Allied Steam Motors*, by W. H. Uhland. A series of articles on Corliss valve-gears have also appeared in *The Practical Engineer*, 1893-4.

Actuation of Corliss exhaust-valves.—In Corliss engines the exhaust-valves are worked continuously, and

without detachment. As in steam admission-valves, a wrist-plate may or may not be adopted. When a wrist-plate is adopted, the same advantages may be secured, though they are here less valuable than in connection with steam admission-valves. Compression may be adopted to any desirable extent, especially as the clearance spaces are small, so that compression may be commenced at a later stage than would be necessary to secure the same degree of compression in larger clearance spaces.

Application of Corliss valves.—A considerable degree of similarity exists between the steam distribution effected by Corliss and double-beat valves, when everything is in good order, as is evidenced by the indicator diagrams. But important differences exist in the means whereby such results are secured. In the closing of Corliss valves of any kind no trace of a blow exists, as the action is purely one of sliding, and no sudden contact is made with a stop or valve seating. Each Corliss admission-valve occupies a bored chamber, so arranged as to give a minimum of clearance space; the clearance against the exhaust-valves is greater, but still is of moderate amount. With a view to minimize the loss by reason of clearance against the exhaust-valves, these have recently—especially in continental practice—been placed to project into the cavity of the cylinder, in such a manner that a certain amount of space is occupied alternately by part of the piston and part of the exhaust-valve. The construction of Corliss valves is effected with greater facility and certainty than that of double-beat valves. A moderate amount of wear is allowed for by leaving the valves loose upon the spindles, which are made of a flat section, or otherwise prepared to transmit a positive motion. By this means also each steam-valve is free

to leave the face if required, and thus allow the escape of a moderate amount of water, on the rare occasions of its accumulation in the cylinder. The arrangement with steam-valves above and exhaust-valves below is, however, especially conducive to a dry cylinder. The edges of the disengaging detents are liable to wear, but when well designed they may be easily removed at a trifling cost. For this purpose spare pieces should be always kept in stock, ready to change at any time. These points should be freely lubricated. In Ramsbottom's motion, a toggle or knee-joint is arranged to trip at the required point in the stroke, by which means point contact is avoided.

Limits of application of detaching motions.—Valves which close by disengagement are unfitted for a greater speed than about 100 revolutions per minute, on account of the excessive wear and noise which would accompany such use. It therefore follows that a high piston speed can only be secured by the adoption of long strokes when using Corliss or double-beat valves.

Similarity between Corliss and plain slide-valves.—A Corliss valve actuated by the pure motion of an eccentric is subject to the same rules as a plain slide-valve affecting lap, travel, and cut-off. These are, however, obviously modified by the application of a wrist-plate, and still more so by the detaching motion.

Adjustment by governor.—With well-arranged valves and gear of either Corliss or double-beat type, comparatively little force may require to be applied by the governor in regulating the engine. But in some cases, and not necessarily imperfect ones, a considerable force is necessary for this purpose, though only for momentary application. A test as to this condition may be made by a careful application of a spring-balance to the governor-rod when the engine is at work. By ensuring

the application of a light load upon the valve-rods, the dimensions of the detaching surfaces and connections may be reduced to a minimum.

Continuous motion of Corliss valves.—Recently, modifications of the Corliss valve motion have been introduced, in which the detaching motion is dispensed with, and the whole actuated in a positive manner. In such cases the steam-valves are actuated by two positive motions, of which one is adjustable, according to the degree of expansion required. If this principle should prove successful, there is every probability that it will lead to the application of Corliss valves to engines running at a high speed of revolution. Examples of this class are described in *Engineering*, vol. liii., p. 791, and liv., p. 264.

Wheelock valves.—The Wheelock valve motion is worked in a manner very similar to the Corliss gear. Separate steam admission- and exhaust-valves are used, of the gridiron type, the latter being worked continuously, and the former by a dash-pot and trip motion.

CHAPTER XXIV.

CONSTRUCTION OF THE CYLINDERS OF STEAM-ENGINES.

Uniformity of substance and temperature.—A cylinder should be sufficiently stiff and free from original strain to maintain its shape under the action of gravity and the stresses imposed upon it in work. To some extent this is a question of thickness of material, but still more of uniformity of substance. The most perfect arrangement of cylinder in this respect is in the form of a plain pipe, with nothing attached to it but the flanges at each end for securing the respective valve-boxes and covers, and for fixing the whole to the foundation. Such a cylinder is, in use, exposed to one uniform condition of temperature all over, and being of uniform substance it expands equally in every direction. It is also free from the distorting effect of various attachments, which are often applied to a cylinder on one side only. The uniformity and simplicity of such a cylinder give it a great advantage in soundness of casting and perfection of surface, especially in the bore. The cost of the cylinder itself is very much reduced by the adoption of the simple design referred to, but owing to the necessity for the provision in separate form of the details usually attached to the cylinder casting, the total cost of the engine is somewhat increased. The temperature of the body of

an unjacketed cylinder in work is lower than that of the steam supplied to it, and greater than that of the steam exhausted from it. Therefore a valve-chest which applies hot steam to one side of a cylinder causes a distortion of its rectilinear condition. An exhaust-belt or passage has an opposite effect. In many cases the exhaust-belt is placed between the cylinder and the valve-chest, and so exercises a cooling effect upon the steam on each side, which seriously militates against economical working of the engine. The exhaust passage is often continued as a belt partially around the cylinder, and thereby increases the prejudicial effects. In a single-cylinder engine the heat thus abstracted from the live steam is carried away in the exhaust steam and totally lost. In a compound or triple expansion engine this identical heat may be utilized in succeeding cylinders, but a similar loss is incurred in each cylinder which is arranged in the same way, so that no ultimate gain can be definitely said to be secured. Thurston's pickling process may prove useful in this connection.

Use of a cylinder liner.—The complete fulfilment of the condition that the cylinder shall be free from distorting influence is seldom possible, though in a very large number of cases great improvement is to be secured with little trouble or difficulty. But the chief objects referred to are achieved by the use of a cylinder liner. In vertical marine engines, where liners are very largely used, the lower end of the liner is secured by a flange to the cylinder end, and the upper end is pressed or slightly shrunk in, and also made tight by a packing of asbestos or copper wire. This is a good arrangement, but a difficulty is encountered at the lower port, which must be continued beyond the flange of the liner, so as to clear the joint, whereby the amount of clearance at the lower end of the port becomes

appreciably increased. Figs. 48 and 49 show the upper and lower ends of such a cylinder. In horizontal engines and others arranged so that the liner is freely accessible at both ends, the arrangement shown in Fig. 48 may be adopted at both ends. A liner of either

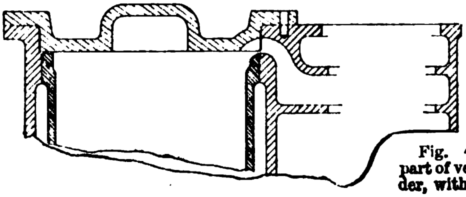

Fig. 48. — Upper part of vertical cylinder, with liner.

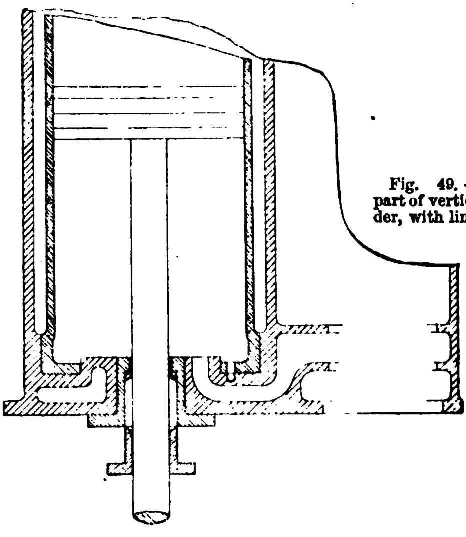

Fig. 49. — Lower part of vertical cylinder, with liner.

kind is not subject to distortion by heat, and may be made of metal of sufficient hardness to assume a high polish in use, and work with extremely little loss of substance by wear. When a liner is used, the cylinder can be made of tough, soft cast-iron, which is much superior for the purpose in every respect, except that

it fails to produce a smooth or durable surface in the bore. The space between the jacket and the cylinder body may be filled with air, or it may be utilized as an efficient steam-jacket. In *Engineering* of July 13, 1894, a cylinder is shown in which Corliss valves are used, and also a liner of which the outer surface is corrugated to increase the amount of surface for the reception of heat from the jacket steam. The two valves at one end are placed in chambers cast along with the cylinder liner, and those at the other end in chambers cast in the cylinder shell. The manner in which the two are connected by means of only one inside joint is ingenious, and appears likely to prove most efficient in use.

Liner cast with cylinder.—Small cylinders are sometimes cast with the jacket cored out of one solid casting, so that the joint between the two is abolished, and occasional trouble of leakage is avoided. The whole must, however, be made of the same metal, the space is difficult to clean out, some risk of unequal thickness in the casting is incurred, and more uncertainty as to the quality of the casting exists.

Damage arising from injudicious application of steam to jacket.—In connection with the use of a jacketed cylinder of any kind, grave risk is incurred by admitting steam to the cylinder and not simultaneously admitting it to the jacket. The same result follows upon sudden admission of full steam to either or both. Calculations —which need not be here fully treated—show in such cases expansion in excess in the cylinder liner, which will fully account for all the trouble experienced by way of started joints. Leakage at joints is avoided by the adoption of solid jacketed cylinders, but the risk of a cracked cylinder is incurred, which is of a far more serious nature. In either case trouble which arises in

this way must often be attributed to culpable carelessness, and should be treated accordingly. In well-designed and constructed work, in which accumulations of water are prevented, leakages from other causes are very rare.

Strength of cylinder.—A cylinder which possesses sufficient substance for stiffness and stability is more than amply sufficient in strength to resist the internal pressure of steam. It is very rare that the circumferential stress due to internal pressure exceeds three-quarters of a ton per square inch, which is within the safe load for cast-iron.

Areas of ports.—The ports of a cylinder are required to provide such an amount of sectional area as will prevent the velocity of steam from exceeding an amount which varies with the pressure, as explained in another chapter. On the other hand, they should not in any case be made much larger than is necessary to fulfil this condition. When ordinary slide-valves are adopted, the sectional area of steam-ports is often made equal to one-twelfth to one-sixteenth part of the area of piston. These proportions have been adopted for all pressures, and for speeds up to 450 feet per minute, giving an average velocity of steam through the port of 90 to 120 feet per second, and a maximum velocity very much greater. For higher speeds, ports of such area are quite inadequate. The dimensions of the port rule those of the slide-valve. Ordinary slide-valves are subject to great frictional resistance, which is another reason why they should be kept as small as practicable. Valves which close by trip motion or otherwise, after remaining fully open until the latest possible moment, allow of some reduction in the area of ports. The transverse width of ports is from two-thirds the diameter of cylinder bore to full diameter, or in some cases a little greater.

Position of ports.—The exhaust-ports should not be placed above the cylinder when it can be avoided. When ordinary slide-valves are adopted, which regulate both steam admission and exhaust, they are usually placed on the side of the cylinder. In locomotive engines the slide-valves are often placed above the cylinders for special reasons, of which the chief is the want of space necessary for other arrangements. When the slide-valves or exhaust-valves are placed over the cylinders, and water in a liquid condition is formed in the cylinder, in quantity just insufficient to open the escape-valves, there is no possibility of its removal, so that it remains and leads to great loss by cylinder condensation. In a less degree the same condition exists when the ports are on the side of the cylinder, and stand above the level of the bottom of the cylinder.

Strength and design of ports.—The metal of the cylinder surrounding the ports where they enter upon the bore cannot be favourably disposed for strength. In some cases a rib is cast along the port, dividing a wide port into two narrow ones, over a considerable portion of the length. This, however, interposes an undesirable amount of resistance to the passage of steam. In all cases the angles should be rounded to a good radius, which should be continued throughout the whole length, and to each termination of each port. Each extremity of each steam-port should, as a rule, be worked to a complete semi-circle, and each angle of an exhaust-port be made to the same radius, as in Fig. 50. The work is thus very much stronger than when the ports are left quite square. The slide-valves and the facings do not wear to a ledge opposite to each angle, and they remain steam-tight longer. All ports should possess a smooth surface, and have the ends chipped and filed fair. Usually the cylinder is continued two or

three inches beyond the inner termination of the port, for the sake of strength, and to provide metal for the insertion of studs for securing the cylinder covers.

Design of cylinder covers.—Each end of the bore of the cylinder is closed by a circular cover, secured by bolts or studs and nuts. Bolts with heads are to be preferred where they can be applied, their chief advantage being the facility with which, as a rule, they can be replaced when broken. In way of the ports and in other similar positions studs are screwed into the cylinder, and nuts are used precisely as for bolts, so that the outward appearance of the whole is uniform. The bridge of metal outside the port should be of such

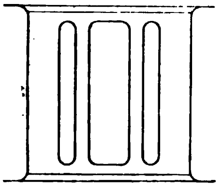

Fig. 50.—Filleting of cylinder-ports.

thickness as to avoid the necessity for drilling the stud-holes nearly through into the port; otherwise the steam exercises a corrosive power upon the stud and the screwed thread of the cast-iron. The general adoption of studs all around the cover would simplify the work, and slightly reduce the diameter of the cover without any important sacrifice in efficiency. But an unbroken surface, $1\frac{1}{4}$ to $1\frac{1}{2}$ inches wide, should be left inside the bolts for making a tight joint. It may be noticed that the use of a cylinder liner causes an appreciable increase in the diameter of cover. A close joint to the edge of the cover presents a neater or more generally approved appearance. But a comparatively narrow joint, as described in the chapter on joint-

making, is superior. As a rule, the bolts should be so disposed that the distance from centre to centre does not exceed 5 inches for bolts of 1 inch diameter, or 6 inches for bolts of $1\frac{1}{4}$ inches diameter. These figures apply chiefly to engines of ordinary dimensions, in which steam up to 150 pounds pressure is used. The pressure of steam is, however, of minor importance, as the bolts are liable to be more severely tested by the occasional occurrence of water in the cylinder. The diameter of bolts, as against steam-pressure, may be more accurately obtained, or exceptional cases considered in the manner given in the chapter on pipes. Under the conditions given above, the thickness of flange of a cast-iron cover will be sufficient if made one-third to one-half greater than the diameter of the bolt, though absolute inflexibilty of flange will not and cannot be secured. A plate, such as a cylinder cover, supported only at the edges, should be sufficiently thick to prevent the imposition of a stress whose maximum intensity exceeds $1\frac{3}{4}$ tons per square inch. The approximate thickness of such a plate of circular form of moderate size =

$$\sqrt{\frac{\left(\begin{array}{c}\text{Diameter of circle of}\\ \text{bolt-holes in inches}\end{array}\right)^2 \times \left(\begin{array}{c}\text{maximum pressure above atmo-}\\ \text{sphere in pounds per sq. in.}\end{array}\right)}{18{,}000}}$$

When the thickness according to this rule exceeds two inches, the cover should be made double, strengthened by internal ribs. Steel covers are adopted in special cases, and are usually of reduced thickness as compared with cast-iron. This leads to greater flexibility and necessity for a re-arrangement of bolts to give closer spacing, so as to secure a steam-tight joint. Unless quite unavoidable, no bolt should be situated in the central vertical plane of a horizontal cylinder. The upper of the two positions thus described is suitable for

receiving an eye-bolt for carrying the cylinder cover, and a bolt in the lower position may interfere with a drain-cock, either in the cover or behind the flange of the cylinder. It is also better—though not so imperative—that the bolts should avoid the horizontal plane. The same principle may be extended to vertical engines. Bolts distributed in this manner, besides presenting various points of convenience, will be found to give a better appearance.

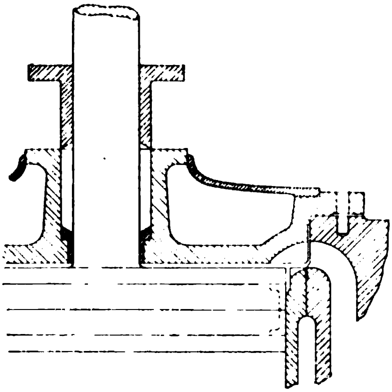

Fig. 51.—Cover for lined cylinder, with stuffing-box.

Each cover should give a definite amount of clearance, as measured from the piston when at the end of the stroke. By the wear of the connecting-rod brasses, the clearance at the end of the cylinder nearest to the crank tends to become reduced, and that at the opposite end correspondingly increased. For this reason the clearance is generally made rather greater at the crank end. The piston should not work past the port so as to obstruct the passage, and the clearance between the piston and the cover must be small. The cover is therefore spigoted into the bore, by an amount about equal to or a little more than the thickness of

metal bridge outside the port. A cover which carries a piston-rod stuffing-box must be accurately centred with respect to the cylinder bore, and must therefore be accurately turned to fit. But the part so fitted need not reach more than three-quarters of an inch below the surface, or the cover would be difficult to remove when choked with dried oil or rust. The remaining portion is turned so as to stand quite clear, as in Fig. 51. The part of the bore of an unlined cylinder into which the cylinder cover fits should remain unaffected, when the cylinder is subsequently re-bored, and is hence made originally of a diameter rather larger than the working bore, the two being joined by a short taper, as in Fig. 52. A cylinder thus treated is said to be "counter-

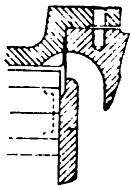

Fig. 52.—Joint of cover for cylinder, unprovided with liner.

bored" or "bell-mouthed." A similar difference of smaller amount may with advantage be adopted when a liner is fitted. Whether the cylinder be fitted with a liner or not, the parallel working bore should be made of length to correspond to the stroke and to the arrangement of piston-rings. When wide piston-rings are used, the length of working bore should be such that at each end of the stroke the outer ring works a little past, and thereby prevents the formation of shoulders. Narrow rings should work as nearly as possible fair with the end of the parallel bore.

Polish of surfaces and use of cover-plates.—The cylinder covers, stud-points, and nuts are generally polished

and maintained in a bright condition. In some cases the central portion of each cover is left black and closed by a thin cast-iron screen-plate, as in Fig. 51, which is polished, so that the entire visible surface is bright. The space between the cover and the screen-plate may with advantage be filled with non-conducting composition. The screen-plate is sometimes carried outside the nuts which secure the cylinder cover.

Valve-chest covers.—The valve-chest covers are generally plain rectangular plates, thickest near the edges, at which parts they are planed on front and back. They are fitted inside to the valve-chest flanges and polished on the outside and edges. An unsupported plate should be so thick as to prevent the imposition of a maximum intensity of stress exceeding $1\frac{3}{4}$ tons per square inch. The approximate thickness of a plate of nearly square form =

$$\sqrt{\frac{\text{Distance of lines of bolt-holes in length} \times \text{do. in width, each in inches} \times \text{maximum pres. above atmosphere in lbs. per sq. in.}}{15{,}000}}$$

If by this rule the thickness becomes excessive, the cover must be strengthened by ribs, which should be placed inside. Such ribs are usually placed outside, where they are under a serious disadvantage as to strength. In exceptional cases, these covers may, for the sake of strength, be made double, in the same way as the end-covers of the cylinders. Valve-chest covers would be partially relieved of strain by reducing their size. But the cover opposite each valve-facing should present a clear opening sufficiently large to allow efficient planing of the face by means of a good machine. Generally, a length of opening three inches greater than that of the facing, and a width of two inches greater, will suffice with care; but if possible more should be allowed. If the planing cannot be finished

in one operation it is indifferently performed, entailing more hand-work in finishing, and also a less perfect result. The covers are held by bolts and studs, generally similar in size and arrangement to those of the cylinder end-covers. These also should avoid the vertical and horizontal centre lines, both for appearance and efficiency. Bolts should not be placed in the corners of the covers, but as in Fig. 53, giving about equal area of contact surface to each bolt. Bolts should never be used in situations in which the heads are exposed to steam, as under such a condition leakage of steam is liable to develop along the bolts. By the exclusive use of well-fitted studs, thoroughly good work is secured,

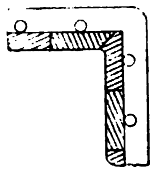

Fig. 53.—Placing of bolts in rectangular cover.

and the size of covers may be reduced to a minimum, or the proportions modified.

Stuffing-boxes for soft packing.—Stuffing-boxes for piston-rods should be sufficiently wide and deep to take a good charge of packing. By this means sufficient elasticity will be secured, and constant attention will not be necessary for the maintenance of a steam-tight condition. When a gland requires constant tightening, it is because the packing is wearing away or becoming more consolidated, and this means friction and loss of power. This may arise from want of truth in the piston-rod, which is obviously best met by restoring the true condition by skimming in the lathe. This, however, cannot be frequently performed, and an appreciable

amount of error may be met by the provision of sufficient elasticity in the packing, by which means also the frictional resistance and the wear of the rod and of the packing are reduced. The length of time which may be run with a high-pressure gland showing no more than a faintly-visible breath of steam is evidence as to the efficiency of the work, and of the attention which it receives. A stuffing-box which is exposed to steam of pressure below that of the atmosphere may become leaky and remain undetected a long time, as steam will not escape. In some cases a puff of steam may occur at the moment of admission, and hence such a puff in a low-pressure cylinder should receive immediate attention. In such a case, or if the vacuum is found to be defective, a fine jet of hot water may be applied to the piston-rod, close against the stuffing-box on the out-stroke. If any serious leakage is taking place, it will be quickly revealed by the sound.

The depth of stuffing-box should suffice to hold the steam; any greater depth causes useless friction. The width should suffice for drawing the packing and inserting new packing in an orderly manner. If ideal conditions could be secured, the depth of packing in contact with the rod would vary with the pressure to be resisted. It is, however, found practically desirable to provide a deeper box for a larger rod, chiefly for the reason that larger rods are usually worked at a higher speed; and a comparatively loose condition of the packing, such as is possible in a deep box, is conducive to its durability. Large stuffing-boxes are, however, often made much deeper than there is any necessity for. On the other hand, small ones are very often made much too shallow. Chiefly for this reason, in the majority of engines fitted with slide-valves, the valve-spindle stuffing-boxes are distinctly more leaky than those of the piston-rods.

Both make the same number of strokes, but the piston-rods move with a much higher velocity. The steam-pressure in the cylinder, however, fluctuates much more than that in the valve-chest, which is to the advantage of the piston-rod stuffing-box. It would, however, be highly inconvenient to make glands for the same size of rod to a greater number of depths than at present, and hence for practical convenience stuffing-boxes will probably continue to be made to vary, chiefly according to the diameter of the rod or spindle. A smooth, parallel piston-rod, truly guided, will work satisfactorily in a stuffing-box packed with asbestos, when the minimum diameter of the box $= (1\frac{1}{8} \times \text{diameter of rod}) + 1$ inch, and the minimum clear depth of packing $= (\frac{1}{2}\ \text{diameter of rod}) + 2\frac{1}{2}$ inches. These dimensions should, however, usually be exceeded. A stuffing-box for a revolving valve-spindle, a stop-valve, or a throttle-valve, in each of which the movement of the spindle is comparatively small, may be made of much smaller dimensions, but the practice is not to be recommended. All stuffing-boxes of small size should be fitted with solid glands of gun-metal, and those of larger sizes with glands lined with gun-metal; all should also be provided with gun-metal bushes in the bottom of the box. Stuffing-boxes should be uniformly packed with separate rounds of packing, and screwed up equally all round. For this purpose the gland is sometimes screwed, but in large sizes it is awkward to use. The operation is, however, most efficiently performed by means of three gland-pins, a rule or gauge being used to ensure exact parallelism. Where this attention cannot be given, two gland-pins should be used for small and moderate sizes, on account of somewhat diminished liability to become twisted in screwing up. Backing-nuts are sometimes provided for use in withdrawing the gland, but they are

unnecessary when the gland is well fitted and receives reasonable attention. Vegetable packing should not be used in stuffing-boxes exposed to steam of a pressure above 50 pounds per square inch, or a temperature above 300° F., on account of its liability to decomposition. Such packings of good quality may, however, be used in other stuffing-boxes of high-pressure engines if desired.

Stuffing-boxes for metallic packing.—Metallic packing for stuffing-boxes is now becoming largely used. Many patterns are in the market, which can be attached to an ordinary stuffing-box without any alteration being made to the latter. But such packings should never be applied to a worn rod without its being accurately turned to uniform diameter, and also adjusted to perfect line, if even the slightest amount of deviation can be perceived when at work. These precautions should never be neglected, though some arrangements of this class possess an undoubted power of yielding slightly to the inaccuracies of the rod.

Cylinder covers directly connected to engine framing. In many horizontal engines of modern design the cylinder cover, next to the crank-shaft, is combined with the bed-plate, with a view to obtain a more direct and efficient connection to withstand the alternate thrust and pull between the cylinder and the crank. In vertical inverted engines, the same object is attained by the most convenient means which are available for connecting the cylinders to the cranks. The standards are, however, seldom cast with the lower cylinder cover, but are attached by means of separate flanges cast upon the cover. The lower cover is attached to the cylinder as usual by bolts or studs. The standards are attached to the cover by similar joints well fitted over the whole surface, to prevent movement. An excellent arrange-

ment is also secured by inserting an entablature or staging plate between the standards and cylinders, arranged to project all round.

Cylinders secured by feet.—Cylinders of horizontal engines which are secured upon a flat bed-plate should have the planed surfaces of the feet arranged as near to the centre plane as possible, to reduce the bending strain. If for long strokes, it is better that feet should be prepared at each end of the cylinder, both pairs being equally well bolted, but the pair nearest to the crank-shaft should be secured against slipping by snugs on the bed-plate and keys tightly fitted in the spaces, by which means the expansion of the cylinder and the piston-rod by heat, in working, will tend to correct each other.

Pipe connections.—The supply and exhaust steam-pipes for each cylinder should be attached to branches suitably placed for the purpose, chiefly with a view to the most direct line possible under the conditions. They are usually placed below the floor of the engine-room, to be out of sight, but this often entails unnecessary bends, and difficulty of access in case of leakage. But with a little care in arrangement pipes may often be carried overhead with great advantage and without offence. In either case, means should be provided for the prompt removal of condensed water by means of traps. The diameters should be sufficiently large to satisfy the conditions set forth in the chapter on pipes. Intermediate-pipes often require some increase in diameter, for the purpose of providing sufficient receiver capacity.

Valves and cocks.—Auxiliary valves are usually provided, to admit steam to the piston at either end of a cylinder, when access by the main valve is closed. These are at all times a convenience in starting the engine by

steam, and are invaluable for warming the cylinders. Drain-cocks are necessary, and they should be fitted with pipes to lead into the open air, where the ends are always visible. Escape-valves of ample area should be provided to each end of each cylinder and to each valve-chest, to be loaded as nearly to the maximum working pressure as possible, and to be provided with convenient easing gear. Weight-loaded valves are more easily adjusted to the proper load than spring-loaded ones. The latter, however, yield more promptly to pressure. Weight-loaded valves with a good spring interposed between the weight and the lever are excellent. Indicator-cocks should be fitted with pipes as short and direct as possible, and with ample bore as smooth and uniform as possible.

CHAPTER XXV.

PISTONS AND PISTON-RODS.

Purpose of piston.—A piston is used to form a barrier across a cylinder, sufficiently closely fitted against the surface of the cylinder to prevent the leakage of steam, and yet sufficiently free to move along the cylinder without encountering an excessive amount of frictional resistance. In all engines which have attained to great practical success, each piston is circular in profile and fits a bored cylinder. The strength of a piston must be sufficiently great to withstand the whole force, due to the application of steam-pressure alternately on the back and the face. The fit upon the piston-rod must be very good, to ensure safety, the absence of movement—due to alternations of force; and tightness—to prevent the insinuation of steam to the fitted surfaces; the workmanship must therefore be accurate throughout.

Steam-tightness along circumference.—By the exercise of very great care and skill, a turned piston and a bored cylinder may be made to fit each other with such accuracy as to remain visibly steam-tight in any position, and yet allow the free movement of the piston. If the fit is very little less perfect, the piston may be made visibly steam-tight by turning three or four grooves in the circumference. These grooves should be from five-

sixteenths to half-an-inch in width, and a little greater in depth, with the edges left quite square and smooth. Such grooves oppose a resistance to the flow of any fluid past them, which resistance arises from the abrupt changes of section presented. Though a piston may be tight in this way when new, it is liable to wear in use, even if working vertically, and therefore carried by outside bearings, but still more so if working horizontally, so that its weight is imposed upon one side of the bore. Changes in temperature also cause changes in the dimensions of the piston and cylinder. Differences in metal also cause differences in expansion, in addition to those due to variations in temperature, as between the piston and the cylinder. The maintenance of a perfectly steam-tight condition in a solid piston is therefore a task of some difficulty, and various types of elastic packing are adopted, to follow the variations in the cylinder, whether regular or irregular.

Piston packings.—For steam of very low pressure, hemp or flax yarn may be used, packed in the piston between two flanges, of which one, called the junk-ring, is removable, and secured by bolts or screws. But vegetable fibres, though elastic and efficient for a time, require early replacement; and when steam of high pressure is used, the corresponding temperature causes the rapid destruction of all such materials. Rope or yarn-packed pistons are, however, still very successfully used for pumps, to pass water at all pressures. Metallic packings are now used exclusively for steam pistons, working at all pressures and temperatures. These are in all cases continuous throughout the circumferential surface except at one point, where a loose joint is made to allow the required elastic movement.

Ramsbottom's piston.—Ramsbottom piston-rings are made to a truly or approximately rectangular section

of small dimensions, so that they fit such grooves as already referred to in connection with a plain, unpacked piston. Usually they are made of steel, but in many cases they are made a little larger in size and of cast-iron, which works better with the cylinder surface. In the former case each ring is made of uniform section throughout, for facility in manufacture. In the latter case the width of face is made uniform, to fit a groove of uniform width, but the depth is reduced at the two ends abutting against each other, by which means the strength of the ring is approximately adjusted to the stress to be resisted. The gap at the junction is closed against a direct leakage of steam by means of a half-lap joint, of about five-eighths of an inch in length, or sufficient to allow for a fair amount of wear. The rings are carefully bent—or turned in the lathe—to such a curve that, when placed in the cylinder, they press against the bore with sufficient force to make a steam-tight joint, and so that each part of each ring shall present, as nearly as possible, uniform pressure against the cylinder bore. Notwithstanding every care which may be bestowed upon the joint, there is often a slight leakage at this part. Consequently three rings are usually adopted for each piston, and means often adopted to shift the joints as far apart as possible. This is done by means of stop-pins, which in order to avoid weakening the rings should be placed at the joints. The full strength of the ring should never be interfered with near to the centre of the length. With moderate care in fitting and in use, these rings never break; but when, from neglect or indiscretion, this result is brought about, the cylinder is sure to suffer great damage. These rings are usually applied in separate grooves turned in the solid piston. Sometimes, in cases in which the steam is very dirty, and

likely to leave such deposit as to interfere with the free working of the rings, the grooves are made of sufficient width to receive two rings each. The ordinary steel rings are easily passed over a solid piston. But when cast-iron rings are adopted, the piston is usually made with a junk-ring. This is also necessary in every case in which the withdrawal of the piston for the inspection, removal, or replacement of the rings is impossible or inconvenient. In the latter cases a junk-ring and packing-blocks are adopted. Ramsbottom rings of steel often fail to receive the careful fitting which they ought to do. This is shown by their condition upon removal after a period of work. They ought to appear uniformly bright on the face, and uniformly worn in thickness. Instead of this, in some cases one part appears abnormally bright, and another part dull black; the latter probably retains its original surface, while the former may be very much worn. A piston in this condition must always cause much waste of steam.

Various metallic pistons.—Some of the early pistons with metallic wearing surfaces were made with one wide ring, kept in close contact with the cylinder bore by various arrangements of hemp packing or metallic springs, the latter of which were usually of the leaf type, and were adjusted by means of screws. In others, two rings were adopted, of half width, with the joints spaced about 90° apart, each spread by a V block, forced into the joint by spring pressure. The latter arrangement gave exceedingly good results, so long as it was in good order, and the cylinder smooth and parallel. But it was found to be deficient in elasticity, and apt to become loose, leaky, and noisy.

Mather and Platt's pistons.—Mather and Platt's piston may be taken as an example of a very popular class, in which two angle-rings of cast-iron, turned all over, are

employed, the combined face width of which fills the space between the solid flange of the piston and the loose junk-ring, as in Fig. 54. Each ring bears against the bore of the cylinder and the flange or junk-ring of the piston, making a steam-tight fit when in good condition. The ring which at the moment is farthest from the pressure side of the piston is the effective one, so that the two are in alternate action. The rings are turned a little larger than the bore of the cylinder, so as to give a certain amount of pressure against the bore. The pressure thus obtained is, however, of small amount, and a spiral spring of rectangular section is used to give increased pressure against the cylinder. The spiral

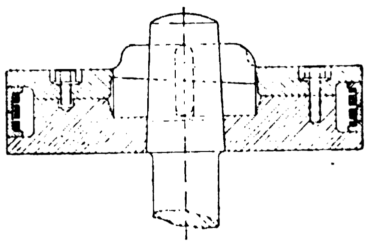

Fig. 54.—Mather and Platt's piston.

spring also tends to spread the two rings apart, so that pressure is exerted against the flange and the junk-ring, and a steam-tight condition is produced. The springs were originally cut from special castings of iron, afterwards of steel, and at present they are usually cast in steel ready for use. The joints of the rings are cut diagonally, to avoid scoring the cylinder, and covered by a tongue-piece to prevent leakage. Sometimes the tongue is riveted to one part of the ring, so that it cannot accidentally drop out. When in good order, this is an excellent piston. It is, however, apt to become weaker by use and wear, and possesses no means of adjustment. In placing this packing in the cylinder,

it is best to withdraw the piston if possible, then insert the parts in their proper order, between the junk-ring and the flange of the piston. Clips are used to compress the packing to such a diameter that the cylinder will receive it. A little rough paper is inserted between the packing and the junk-ring at several places, the junk-ring screwed up so tightly as to seize the packing, and the clips removed. The piston and the packing may then be placed in the cylinder together. When the packing is in position the junk-ring is removed, the paper withdrawn, and the junk-ring cleaned, replaced, and screwed quite tightly by all bolts, each bolt being tested two or three times in continuous order. Throughout this operation, the most scrupulous care should be observed to prevent the access of any trace of grit or dirt of any kind. For this reason the floor and all adjacent parts of the engine should be cleaned before opening the cylinder; these and the piston, and all other parts opened, should be also thoroughly cleaned before closing up.

Buckley's piston.—In Buckley's piston, casing-rings are adopted in contact with the cylinder, equivalent to those last described, but differently formed. The spring used is a flattened spiral coil of steel wire, compressed in length, bent round to a circle, and inserted between the two rings. The spring imposes pressure upon the rings, in its effort to increase its length. The pressure is not communicated to the rings in a radial direction, but to a part of each which is cut to a conical or oblique surface, which is disposed at such an angle as to distribute the pressure in the most suitable proportion between the bore and the piston flange. This piston, shown in Fig. 55, is practically as good as the last, and possesses the advantage of adjustability by means of loose rings, rove on a flat

bar which is inserted in the adjoining ends of the spiral spring. This piston-packing is more easily put together than the last described. The same precautions as to cleanliness and complete screwing up should be fully observed.

The pistons described may be regarded as typical ones, and have been described to illustrate the principles involved. Other makers adopt the two casing-rings with no essential change, but in combination with springs of various kinds.

Elasticity and means of adjustment.—When a piston has been at work for some time, the casing-rings, and

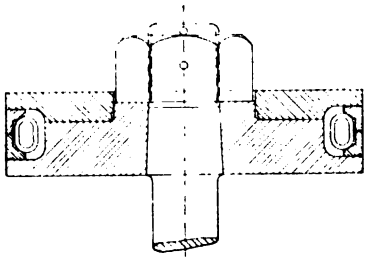

Fig. 55.—Buckley's piston.

possibly the springs, lose some substance by wear. The loss of substance is greatest at the surface in contact with the cylinder. This should be made good by the packing following up. But during this operation the elastic force of the rings and spring becomes spent, unless—as in Buckley's piston—means of adjustment are adopted; and no piston in extensive use possesses automatic means of adjustment. The consequence is that pistons are liable to become leaky, while many must remain so. Many attempts have been made to provide automatic means of adjustment by means of steam-pressure, based upon an impression that the application of steam at cylinder pressure to the back

of the packing would force the rings into uniform contact with the bore. In many, if not in all cases, however, steam at a pressure equal to the working pressure in the cylinder insinuates itself between the ring and the bore, and practically the full pressure is maintained throughout the full depth of the ring. Full steam-pressure is never excluded from the back of the packing by any means ordinarily adopted, so that it cannot be increased by special means of admission. It follows therefore, that, so far as steam-pressure is concerned, the rings are practically in complete equilibrium, and that only the pressure due to their elasticity as metallic structures is effective to keep them in steam-tight contact with the cylinder. It has been lately proposed to recess a portion of the surface of each ring, so as to place such portion in communication with the exhaust side of the piston, and, as it were, to suck the ring into contact with the cylinder.

Metal for rings.—Cast-iron is the best substance of which to make piston-rings for general use. If very soft, the rings wear rapidly. If, however, as hard as the cylinder, there is some risk that the latter will suffer seriously by wear. Rings should therefore be made of metal just sufficiently hard to avoid cutting the cylinder. Piston work is one of the many branches in which cast-iron shows itself to be specially well adapted for acquiring a good surface, and working with little frictional resistance and great durability. Probably its superiority is connected with its crystalline nature. Steel is used in Ramsbottom rings and a few others, on account of its strength in small sections, and the facility with which standard sections may be drawn to form. Its surface is, however, inferior to that of cast-iron. Brass, or one of the analogous alloys, is often used in cases where the machinery is at rest for such lengthy periods as

would suffice to fasten iron or steel by corrosion. The metal used should, however, not be powerfully electrically negative to iron, or the bored surface of the cylinder will suffer. Brass is quite satisfactory for this purpose while in good condition, but liable to rapid wear.

Junk-ring.—A junk-ring is required for most of the packings in use, and is shown in Fig. 54. It should be secured by set-screws, the heads of which are recessed or countersunk below the surface of the ring. Some years ago these were always made of forged copper or cast gun-metal, and screwed into nuts of gun-metal, which were either sunk into recesses in the piston or screwed into large holes tapped into the piston and afterwards drilled and tapped to fit the set-screws. At that time this arrangement was necessary on account of the corrosive action of the fatty acids introduced into the cylinder as component parts of the tallow used for lubrication. The fitting of copper bolts and gun-metal nuts still remains good practice, but is not imperatively necessary with the use of mineral oils of neutral reaction. The set-screws may therefore now be made of tough Yorkshire iron, and tapped into the solid casting of the piston. They should be of ample length, but should not reach so far through the piston as to involve risk of leakage against the screw-points. If the thickness of the piston casting is insufficient to allow metal to ensure tightness below the set-screws, tightly-fitted studs should be adopted, but these are not desirable. The set-screw heads—or nuts of studs—may be checked from slacking back by means of fitted guards attached by set-screws. The use of these, however, increases the amount of clearance necessary, which is prejudicial to economy; and well-fitted set-screws will hold quite securely without assistance.

All tapped holes in work of such importance as that in question should be tapped truly perpendicularly, and not tapped by hand without guide. In the latter case, even a moderate amount of accuracy is not likely to be secured. Three or four forcing-screws should be provided, for facility in removing the junk-ring. Tapped countersunk holes should be provided to receive the forcing-screws, the holes to be closed by set-screws when out of use. Each set-screw should project through the junk-ring into a recess in the piston, as in Fig. 54, by which means the hole is kept quite clean, and the set-screw is secure against slacking back. The junk-ring should be ground to a steam-tight fit upon the piston; otherwise steam will pass alternatively in each direction

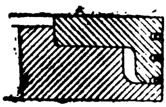

Fig. 56.—Ramsbottom's piston-rings.

with alternate strokes of the piston. The edge of the junk-ring must be truly accurate with that of the piston-flange. This is secured by the provision of a boss upon the piston, turned to fit the bore of the junk-ring. This fit need not be continued over a greater depth than about three-quarters of an inch. By cutting the rest to a clearance, the removal of the junk-ring is very much facilitated.

In many cases Ramsbottom rings have been used in pistons constructed for other types of packing, when the junk-ring has been discarded in favour of one with a projecting part, in which grooves could be cut for the rings, as shown in Fig. 56. This is easily removable without disconnecting the piston, an arrangement which is found to be so convenient that it is often adopted

in new work. Modifications of this arrangement are also adopted for cast-iron rings which cannot be slipped over a solid block.

Clearance space in piston.—The space left behind the rings in a piston should be reduced to its lowest convenient limits, as, unless the rings are exceptionally well fitted, it must be alternately filled with steam and emptied in each stroke, so as to virtually form part of the clearance space.

Weight and strength of piston.—Pistons for stationary engines are usually made solid. Up to a diameter of five feet there is very little advantage to be derived from coring, while the piston is weaker and more troublesome to make. In an engine working expansively, a certain amount of weight is advantageous in absorbing work at the beginning of the stroke when in excess, and restoring it afterwards, when less work is yielded by the steam. Thus, up to a certain point, the pressure upon the crank-pin, due to steam applied expansively, is rendered more uniform by means of an increase in the weight of piston, piston-rod, connecting-rod, &c. Pistons of ordinary diameters in stationary engines always possess sufficient strength, when made solid with convenient widths of packing, junk-ring, and flange, and with the piston of parallel thickness.

Attachment of piston to rod.—The piston should be fitted to the rod by a taper of about one diameter in twelve lengths. This taper will hold the rod securely and steam-tight, and is convenient in every respect. The joint will require some force to part it after a period of work, but will seldom cause serious trouble in this way. A finer taper would be difficult to separate and a coarser one would be liable to allow leakage of steam. This taper will cause the imposition of a considerable bursting stress, if water should be present in the

cylinder when the piston is in rapid motion. Hence the necessity for avoiding excessive reduction of metal by coring. The amount of bursting force, as compared with pressure along the rod, is dealt with in connection with cross-heads. But the force of a blow arising from the presence of water in the cylinder cannot be accurately estimated. The piston may be secured to the rod by means of a cotter made in halves, and prevented from slacking back by means of the junk-ring, as shown in Fig. 54. An alternative plan, in which a nut is adopted, is shown in Fig. 55. When used alone, the cotter is more safe against backing, but the nut can be very efficiently held by a split pin through or in front of it, with or without the adoption of a lock-plate secured by set-screws. The cotter weakens the rod to a greater extent than does the screw. The nut occupies a slightly greater length, but this is seldom any serious objection, and the boss may be turned a little below the surface of the junk-ring, as shown in Fig. 55. In the majority of cases a distinct preference should be accorded to the screwed arrangement. In an overhead-crank or beam-engine the direction of cone is reversed, or the large part placed at the end.

Piston-rods.—Up to a recent period, a piston-rod was usually made from a scrap-iron forging of highest quality. In present practice it is made of soft steel. The latter is in every respect as good as the former; its tensile strength is greater, and it is less liable to wear in ridges, or "flute." It is usually fitted into the cross-head by a taper, the large end of which is rather less in diameter than the body of the rod. In most cases the shoulder thus formed is not allowed to bear against the cross-head, the whole of the pressure being borne upon the taper. The object of this change of diameter is to allow for the re-turning of

the rod without interfering with the fit in the cross-head. The minimum proportions to be adopted for piston-rods should be such that, in ordinary work, no part will be subjected to a stress in tension or compression greater than 3 tons per square inch in steel, or 2½ tons in wrought-iron. This will allow for moderate shocks arising from water in the cylinder or from other causes. In some cases the piston-rod and the cross-head are forged together, when the diameter in the piston must be reduced below that of the body, or split bushes and possibly a split gland adopted in the stuffing-box.

Wear of surfaces.—The wear of a piston, especially that of a horizontal engine, is reduced by allowing ample width of bearing surface. The material of the cylinder is an important point, and, as elsewhere explained, the wisest course is to provide a liner of harder material than can be practically adopted for the complete cylinder. In use this will assume a surface almost vitreous in its nature, and frictional resistance will be minimized. The wear of the piston-rod depends upon the condition of the packing in the stuffing-box. Wear, generally, is affected by the accuracy of fit and motion.

Back piston-rods, or "tail rods."—In many engines, both horizontal and vertical, the piston-rod is continued through the back or upper cover of the cylinder and a stuffing-box provided, for the sake of steadiness. In some cases such a tail-rod is omitted in the low pressure on account of the stuffing-box, and its liability to inward leakage of air, to the prejudice of the vacuum. When, in a horizontal engine, a heavy piston is mounted upon a continuous rod, the rod is often curved upwards, so that its weight and that of the piston will just suffice to bring it to a straight line when supported at each end. When at work, one end is supported by the cross-head, and the other end by a corresponding plain

slide. The whole weight is thus carried upon surfaces which are continuously visible, and may be efficiently lubricated. The pressure imposed by the piston upon the bore of the cylinder is thus quite uniform throughout the circumference. In some cases partial relief is thus sought.

Test upon accuracy of line.—In a horizontal engine of the original type, the accuracy of line of cylinder, piston-rod, and of cross-head motion may be tested by slacking back the front cover, and noting the degree of accuracy with which it fits the counterbore of the cylinder with the piston in various positions. In this test, regard must be paid to the deflection of the piston-rod, especially if of considerable length. For this reason it may be necessary to support the weight of the cover from above during the test. The above test is inapplicable to horizontal engines of which the front cover is not removable, as also to vertical engines. In such cases the inner bush of the piston-rod stuffing-box should be withdrawn, and used exactly as the entire cover in the former case. If the bush is not of convenient diameter a small gauge may be specially made for the purpose.

CHAPTER XXVI.

CROSS-HEADS AND CONNECTING-RODS.

Conical fitting of piston-rods.—The cross-head may be made to fit the piston-rod in a conical surface, tapered in the proportion of one diameter in twelve lengths, as in the piston. The cross-head is, however, not exposed to the insinuating action of steam, and the taper may be made more obtuse—even to the ratio of 1 in 8—if desired. By this means the cross-head will be exposed to a correspondingly reduced total bussting stress. But the diameter of the taper at the large end is usually reduced below that of the body of the piston-rod. A rapid taper continued through the cross-head, as often arranged for cottering, causes serious reduction in the effective section of the small end of the taper to resist the tensile stress imposed upon the rod. If, however, the taper be continued only a short distance into the cross-head, and the piston-rod continued parallel through the remainder of its length, it may be finished with a screw-thread to receive a nut, which will give an excellent arrangement, as shown in Fig. 57. The proportions adopted should be such that the absolute pressure upon the conical surfaces when of steel does not exceed 6 tons per square inch. This should be worked out in detail, making allowance for any

cotter-holes or key-ways, but an approximation may be obtained by allowing 6 tons per square inch upon the difference in area between the cross-section at the large end of the conical bearing surface and that at the small end. That is, the intensity of pressure upon the conical surface may be assumed to be equal to that upon a plane shoulder, whose outer and inner diameters are the same as those of the conical surface, and similarly corrected on account of cotter slots, &c. The conical surface is of greater area, but the pressure upon it arising from the wedging action is also greater in the same proportion, and this is the case with all pro-

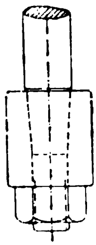

Fig. 57.—Fitting of piston-rod in cross-head.

portions of taper. If the effective length of the boss is exactly equal to that of the taper, the thickness of metal to resist bursting will be quite independent of the length, or will be the same for a long cross-head to suit a rod of fine taper as for a short cross-head to suit a rod of obtuse taper. When cotter-holes are cut through the cross-head, the material opposite should be regarded as possessing no strength or bearing power, and the portions at each end only considered as useful, as shown by cross-hatching in Fig. 58.

Steel piston-rods and steel cross-heads are quite safe under a bearing pressure of 7·5 tons per square inch,

or one-fourth greater than the 6 tons given above. This difference should, however, be allowed as a margin, on account of the initial pressure due to cottering or screwing up. Cast-iron may be allowed to sustain the same pressure as steel, but if either of the surfaces

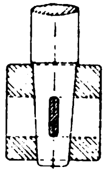

Fig. 58.—Section of cross-head secured by a cotter.

be of wrought-iron, the pressure in ordinary work should not be allowed to exceed 4 tons per square inch. For this purpose the maximum difference in steam-pressure on the two sides of the piston should be adopted.

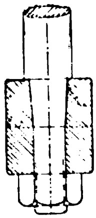

Fig. 59.—Section of cross-head as affecting strength.

Strength of cross-head against bursting.—The thickness of material in the cross-head which is necessary for the purpose of resisting bursting =

$$\frac{\text{Pressure per sq. in. upon conical surface} \times \text{half diam. of taper in inches}}{\text{Tensile working strength of material per sq. in.}}.$$

Assuming that, as in Fig. 59, a shoulder is fitted to a bearing uniform with that of a conical portion, the

pressure per unit area will be the same as on a conical surface of the given taper, and whose larger and smaller diameters are equal to those of the bearing surfaces; or just as though the conical bearing surface was continued in the manner shown by small dots. The pressure per square inch is the same throughout, but the total pressure is less than that which would prevail upon an entire conical surface, in proportion to the area of surface in contact. The great difficulty encountered in securing a fit, uniform on the two parts, and maintaining it under all conditions of work and exceptional strains imposed upon the rod, should not be overlooked. The bursting force applied upon the conical surface is opposed by the surrounding material, which is distinguished in the figure by double cross-hatching. The remainder of the material in the boss is useful in providing transverse strength to oppose the pressure upon the shoulder. The maximum load as against bursting which may be applied to steel of exceptional quality is 7 tons per square inch. Ordinary steel should not be loaded above 6 tons, or say 4½ tons, as calculated from the maximum effective steam-pressure. Similarly, wrought-iron may be loaded to 4 tons, and cast-iron to 1 ton, as calculated upon the pressure applied in each case. Certainly many cast-iron cross-heads are working above the limit of stress given, but they must be made of exceptionally good metal, or by reason of good management have fortunately escaped exposure to such accidents as should always be provided against.

Articulation of connecting-rod.—The cross-head is prepared for attachment to the connecting-rod in such a manner as to allow the vibration of the rod in working. Either the cross-head or the connecting-rod must be "forked," so as to take the stress in two places, equally disposed on each side of the centre line. As

a rule, the cross-head is forked, and the cross-head pin fitted in each side and keyed to prevent motion. In some cases a forked connecting-rod is preferred, but this is more costly to make, and subject to irregularity in cottering or in wear of brasses on the two sides. A forked connecting-rod should therefore be made with strength in each part, considerably greater than corresponding to half the work performed, so that a margin is provided against such irregularity. The cross-head pin is required to be easily removable when a solid-ended connecting-rod is adopted. In other cases also this is a convenience. In all parts, the strength of the cross-head may be calculated upon a tensile working stress of 4 tons per square inch in wrought-iron, and $4\frac{1}{2}$ tons in steel, or shearing stresses one-fourth less.

Cross-head guides.—A cross-head is required to perform the office of guiding the end of the piston-rod in a straight line, notwithstanding the varying angularity of the connecting-rod. The deviating force upon the cross-head in a transverse direction is comparatively small in amount, but must be accurately provided against. The cross-head is prepared with sliding surfaces, or the cross-head or the cross-head pin is prepared to receive separate pieces for this purpose. These may be of steel, but unless hardened they are better if of cast-iron; this also applies to the surfaces with which they work in contact. They should be arranged so that the actual pressure upon the sliding surfaces at any moment will not exceed 40 pounds per square inch upon cast-iron surfaces, or 80 pounds if both surfaces are of hardened steel, ground to a surface, as in the slides of locomotive engines. The calculations upon this point should include the weight of all parts, and the pressure due to the working force imparted by the piston-rod to the connecting-rod. The amount of the latter is

approximately obtained from the mean pressure upon the piston, multiplied by the length of crank—or the half-stroke of the engine—and divided by the length of the connecting-rod; in each case the dimensions to be measured over centres. The exact pressure may be ascertained by calculation, or by the use of the parallelogram of forces. This operation should be repeated for a number of positions in the stroke, of which the point of cut-off is often the most important. Usually the slide-bars of a horizontal engine are provided in duplicate to withstand pressure from above and below. But in ordinary direct driving, with the crank-shaft always revolving in the same direction, the pressure upon the cross-head due to positive steam-pressure in the cylinder is constantly exerted in one direction. In Fig. 60 the

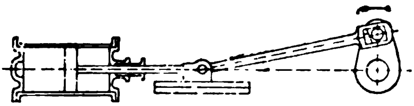

Fig. 60.—Diagram showing rotation of crank-shaft.

crank of a horizontal engine is shown running in the usual direction. The force of compression exerted by the steam upon the piston-rod and connecting-rod throws a downward pressure upon the cross-head. During the return stroke the rods are in tension, and again a downward pressure is imposed upon the cross-head. The cross-head, therefore, exhibits no tendency to rise, and may be worked without any upper slide. On rare occasions this has been done, and in very many cases the surface area of the upper slides is much smaller than that of the lower ones. Conversely, when an engine runs in the opposite direction, the chief pressure is upon the upper slide. But in such a case the weight of the rods and cross-head calls for efficient slide surfaces

below, to prevent knock. When the steam is subjected to compression previous to the termination of the exhaust stroke, there is always a necessity for double slides; and in all cases the event of water in the cylinder should be anticipated, though the probabilities may be against such an occurrence. In marine engines the cross-head slides are arranged for full efficiency with respect to ordinary work. But occasional reversal of the direction of revolution is provided for by reduced area in cross-head slides, in some cases even as little as one-third as much area being provided in the latter case as in the former. This would be quite inadequate, except for the reason that such work is required only for short periods.

The cross-head pin should be placed in the centre of the length of the cross-head block, so that each end of the slides may be loaded with uniformity. In all cases the fixed slides are of greater length than the slide-blocks, or the surfaces attached to the cross-head. The latter should be as long as can be arranged with convenience, as regards surrounding details and freedom from appreciable bending. The length of the fixed slide surface should be such that the slide-block will project beyond it a few inches at each end when at the end of the stroke. This will prevent the formation of ridges at each end of the fixed slide, promote uniformity of wear, maintain a good surface, and facilitate lubrication, which last question is dealt with elsewhere.

Cross-head pin.—The cross-head pin should be made of such proportions as to give bearing surface for the connecting-rod brasses equivalent to a load of 1000 pounds per square inch upon the longitudinal section, or the area obtained by multiplying the length of bearing surface by the diameter. For this purpose, when steam is cut off at one-quarter or one-third of the

stroke, the mean load may be taken, hence the area of the cross-head journal in square inches =

$$\frac{\text{Indicated horse-power upon one crank} \times 18{\cdot}33}{\text{Revolutions per minute} \times \text{length of stroke in feet.}}$$

The load thus allowed upon the cross-head pin is much greater than that upon the crank-pin, as the motion is very limited, and universal experience demonstrates the safety of such a course. The corresponding maximum load in either case is greater than can be applied with equal safety, and with equally perfect lubrication, to bearings of equal dimensions uniformly loaded. This appears to be due to the alternations of the load, by which the entry of the oil for lubrication is much facilitated. With care the loads

Fig. 61.—Section of ordinary cross-head pin after wear.

Fig. 62.—Musgrave's improved cross-head pin.

given may be exceeded 20 to 25 per cent., but this should only be done under exceptional conditions of necessity. An ordinary cross-head pin wears only on the two sides, leaving the pin approximately oval, or rather lenticular in section, as shown in full lines in Fig. 61, when it is quite impossible to keep a tight bearing. Messrs. Musgrave adopt an excellent plan to overcome this defect, which consists in cutting a flat on each side of the pin, and cutting away the brasses to correspond, as in Fig. 62. By this means the effective area of surface exposed to pressure is reduced to a trifling extent, on account of which no allowance is necessary even when the bearing is new. But while in an ordinary case the effect of slight wear is to interfere with the fit of the surfaces, in the improved form the

wear is much more uniform, and the bearing is really safe under a greater load than an ordinary one.

The best material for cross-head pins of short length is wrought-iron, case-hardened and ground to a good surface and true form. These are almost proof against wear. Hard tough brasses working in contact with such pins also remain in good condition a long time. But steel of the quality suitable for heavy shafts may be adopted for long pins, which are often required on account of the necessity for placing the slide-bars some distance apart. Long cross-head pins are sometimes required for driving air-pumps, also for use with parallel motions, and for other purposes. In all such cases the transverse force applied to the pin should be estimated with the utmost practicable accuracy, and the strength of the pin verified as in a shaft.

Variation in speed of piston and length of connecting-rod.—The speed of the crank-shaft of an engine must be maintained within certain limits of uniformity which, though not accurately defined in all cases, are pretty well understood. When the crank stands at right angles to the line of the piston-rod, as in Fig. 60, the obliquity of the connecting-rod causes the piston to stand somewhat nearer to the crank than the centre of the cylinder. Consequently, if the crank-shaft revolves equal amounts in equal times, the piston must move over a greater distance during one-half of the time occupied by a complete stroke than during the other half. The distance through which the piston moves during any period, multiplied by the average pressure during the period, gives the amount of work done. Therefore if any appreciable portion of the work-producing movement is taken from one quarter revolution and added to the other, and repeated inversely during the return stroke, it follows that the amount of work

imparted to the crank-shaft during one half revolution is very different from that imparted during the alternate half revolution. These differences are greater in amount when a short connecting-rod is adopted than with a long one. It has, however, been practically demonstrated that for horizontal stationary engines used for purposes in which steady driving is of importance, a connecting-rod whose length, measured over centres, is three times the length of stroke, satisfactorily fulfils all requirements. Beyond this length the rod would become unwieldy, the bed-plate or frame would assume excessive length, and the engine would occupy an unnecessary amount of space. In vertical engines the length of connecting-rod may be made equal to 2½ times the stroke, while the connecting-rods of marine engines are made still shorter.

Strength of connecting-rod.—A connecting-rod is required to alternately transmit tensile and compressive stress. In addition, there is a transverse force applied to it, by reason of the oscillation of the end of the connecting-rod which is attached to the crank-pin, commonly known as the "large end" of the connecting-rod. The last element is an important one in small and very high speed engines, but not in large engines working at a moderate speed of revolution. Rods of H section are used for small engines and for locomotives, for which they are in every way well adapted. A well-designed rod of hollow circular section would be most efficient, but would be difficult and costly to construct. For large stationary engines the body of the connecting-rod is almost invariably made solid and of circular form, finished in the lathe, for the sake of economy and appearance. For the sake of stiffness the diameter of a connecting-rod is always greater than would be decided upon from a consideration of its

ultimate strength as a strut or a tie, or its transverse strength against bending when driven at a high speed. That of a large horizontal engine is usually made of spindle shape, with the largest diameter near to the centre of the length. As a rule, the largest diameter is about sufficient to give a mean direct stress of 800 to 900 pounds per square inch upon the material, and the smallest diameter a mean direct stress of 1500 to 1600 pounds. In many cases, especially in vertical engines, the largest diameter is arranged at the crank-pin end, and a straight, even taper taken to the opposite end. In other cases the half length—or a little less—nearest

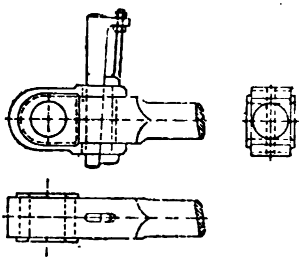

Fig. 63.—Connecting-rod end: strap pattern.

to the crank-pin is turned parallel, and the other half to a straight taper; or the two portions may be joined by a curve.

Strap-ends of connecting-rods.—Each end of a connecting-rod is prepared to rigidly and directly support the inner brass against compressive force, and also, either directly or by means of straps, plates, or bolts, to rigidly support the outer brass against tensile force. The plain strap end of a connecting-rod, as shown in Fig. 63, has been adopted more largely than any other in stationary engines, and over a longer period of time. It is an excellent one for use, but costly to fit properly. In this the connecting-rod is

itself finished with a square portion, through which a slot is cut to receive a cotter and gibs. A square-backed brass is fitted to the end of the rod, and a corresponding one, usually round-backed, fitted opposite. Both brasses are provided with flanges on all sides, and are made thicker at the bed and crown than at the joint, on account of the difference in wear to which they are exposed, and also for the sake of stiffness. A wrought-iron strap is fitted round both brasses, and made sufficiently long to receive cotter-slots matching those in the connecting-rod. The strap is made thicker at the crown than at the sides, chiefly because of the hammering action which is sometimes allowed to develop, and which is apt to cause a distortion of the strap if left throughout of a thickness which suffices at the sides. In way of the cotters the strap is again increased, to compensate for the material removed in the slot, and to give good bearing surface for the cotter. The surface for the latter purpose should not be less than 1 square inch per $2\frac{1}{2}$ tons load, calculated upon the combined area of the two sides of the strap. The strap may be of rectangular section throughout, in which case it is finished all over on a slotting machine, or the inside and the two faces may be finished on a slotting machine, after which the whole is cottered up and the outside finished in a lathe. In many cases the outer brass is prepared to fit the strap with a flat part in the centre, to reduce wear by hammering or chafing. The gibs are arranged with hooked ends, to prevent the strap from opening, and also vibration in work. Sometimes one gib is used, as in the figure, and at other times two, when the cotter is passed between them. One gib is provided with a screwed stem, standing parallel to the surface, in contact with the cotter. The cotter is provided with a drilled lug to fit the

gib-screw for fine adjustment, by means of nuts inside and outside. In many cases a safety bolt is also passed through the whole, parallel to the cotter. In less important work, only a gib and cotter are provided, without any adjusting screw. The adjustment is effected by tapping the ends of the cotter with a hammer, and securing by means of a pinching-screw at one side. In all good work, the cotter-slots are finished with rounded edges, as being much less liable to break at the corners. The cotters are also less liable to become indented in use. The bearing area of round-edged cotters should be calculated upon the width of

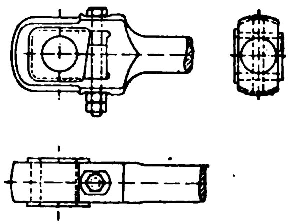

Fig. 64.—Connecting-rod end: Allen pattern.

slot, and not measured around the curve. The slots for round-edged cotters, and the cotters themselves, are generally finished with great accuracy by machine. If this is not done, the same precision of fit may not be secured, in which case square-edged cotters would be preferable.

Solid ends of connecting-rods.—The connecting-rod end shown in Fig. 64 was first brought into notice upon the Allen engine about 1867, and has steadily advanced in favour since that time. The rod is forged with large solid end, through which a rectangular eye is cut, leaving rounded corners. Brasses are inserted, with flanges where possible, but obviously these cannot

extend all round. The bed of the inner brass is cut to fit a taper die, which is adjusted by means of a screw passed through it. The screw is fastened by means of lock nuts, or pinched by a side-screw. The length of die should be not less than two-thirds of the length of opening, and the corners of the die should be left square, so as to obtain the best possible bearing. This type of connecting-rod end is modified from one in which a square-edged cotter bore against the whole length of the brass, but only a small portion of the width. Hard gun-metal should not be loaded with more than 15 cwt. per square inch of surface in still contact with iron. In a modification of Fig. 64 chiefly

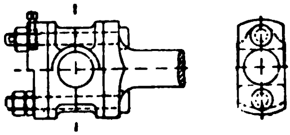

Fig. 65.—Connecting-rod end: marine pattern (two alternatives).

used for locomotives, the outer part is made in the form of a loose strap, secured to the rod by two strong, fitted bolts. The brasses are fitted with flanges all over, as in a strap-end, but their adjustment is treated exactly as in a solid end. When well proportioned, so that the brass is not overloaded, and when well fitted, this is a most excellent but costly arrangement. It is usually adopted with the object of securing great safety at a high speed, and practically the advantages of a solid end with the facility for taking apart, and for application to double-webbed cranks, which are not possessed by the latter.

Marine pattern connecting-rods.—For ordinary use, the marine pattern of connecting-rod end, as in Fig. 65,

possesses most of the advantages attributed to others, and is less costly to make than some of the best ones. The solid rod is provided with a flat palm or table, against which the inner brass is fitted. The outer brass meets this, and a wrought-iron plate is provided outside, to give a good surface against the brass, and avoid the cutting into the brass which would otherwise occur. The plate also keeps the brass rigidly in form and position. Two stout bolts are passed through the whole, the nuts of which are secured by pinching-screws or safety-rings, with or without the addition of lock nuts. These together correspond to the strap in Fig. 63. The thickness of the brasses should be as uniform as possible, with a view to avoid liability to break across from heating and concussion. In very large sizes the brasses become very heavy and costly, when the plan is modified by making the whole end to nearly the same outline in wrought-iron, solid with the rod; this is then bored for a brass bush and for the bolts, machine finished all over, and cut to admit the bush. That shown in Fig. 65 is, however, much better, and should always be adopted if possible. The bushed end would probably be much improved by fitting with H strips in the manner described in connection with main bearings, with the object of preventing the brasses closing upon the pin.

Proportions of crank-pin.—For stationary engines with overhung cranks, the crank-pin bearings may be made up to a length one-third part greater than the diameter. In modern practice crank-pins are largely made up to diameter nearly equal to the length, for the sake of greater stiffness. For double sweep, or two or three-throw crank-shafts, in which the whole of the work is dealt with at one end of the crank-shaft, the crank-pins are of greater diameter than in overhung

cranks, and the length may be made equal to the diameter or even rather less, without incurring deficiency of bearing area.

Direction of wear of brasses.—It will be noticed that the different patterns of connecting-rod vary in the direction in which they allow wear to be taken up. Some cause an increase in the distance of centres, by reason of wear, while others cause a decrease. As it is desirable that the distance of centres should be affected as little as possible, it is well to arrange the opposite ends differently. The arrangement shown in Fig. 64, and the similar one with bolted strap, may be modified so that the wedge may be placed against either the inner or the outer brass, so that the direction may be regulated at pleasure in the original design. Compensation in this respect is, however, only partial in any case, as the crank-pin brasses usually wear more rapidly than the cross-head brasses.

Cross-head fitted with brasses.—The cross-head brasses are sometimes attached to the cross-head itself, when they are generally made according to the modified bolted pattern as last described for connecting-rods. The cross-head pin is then usually of wrought-iron, case-hardened and ground. When this plan is adopted, there is often great temptation to make the solid forked end of the connecting-rod much too light, so that it will always spring and ultimately fail.

Materials for brasses.—The brasses for connecting-rods and cross-heads may be either of solid gun-metal or phosphor bronze for continuous bearing surface; or any metal as a support for strips or blocks of white metal, the whole of which are dealt with in connection with main bearings. White metal may also be used, but cannot be recommended, in connection with the sliding surfaces of cross-head blocks, but should never

be used in connection with cross-head pins or other reciprocating circular surfaces.

Linked parallel motions.—A parallel motion is occasionally adopted instead of slides for guiding the cross-head. In connection with beam-engines this has been almost universal; but it is sometimes adopted in other cases. The simplest motion is shown in Fig. 66, in which *a* is a fixed point, *b* is a point movable along a straight line, which being continued would pass through *a*; *c* is a point which moves in the arc of a circle, around centre *a*; the radius-bar *a c* and the main lever *b d* are constant in length; $ac = cd = bc$;

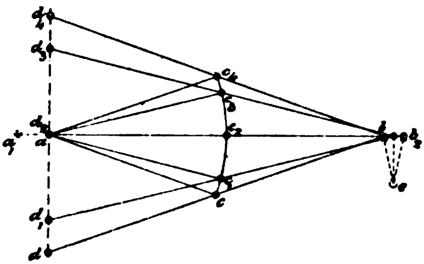

Fig. 66.—Simple parallel motion.

d moves in an absolutely straight line, which traverses point *a*. In some cases the stud which represents point *b* is mounted upon a vibrating link *b e*, which is shown by a dotted line, but of reduced length, to exaggerate the curve described by point *b*. The curvilinear motion of *b* interferes somewhat with the accuracy of motion of point *d*. Parallel motions applied to horizontal and vertical engines generally belong to the type described, but more or less modified according to the conditions of each case.

A second form, applied to beam-engines, is shown in Fig. 67. Point *a* is fixed as before; point *b* represents

the end centre of the beam, and point *f* the main centre; point *e* is midway between points *f* and *b*; the radius-bar *a c* = the parallel-bar *c d* = *b e* = *e f*; the length of the outer links *b d* = that of the inner links *c e*; point *d* will then move in a practically straight line traversing point *a*. If the length of links *b d* and *c c* were indefinitely increased, the accuracy would become absolute. Point *g*, midway between points *c* and *e*, will also move in an approximately

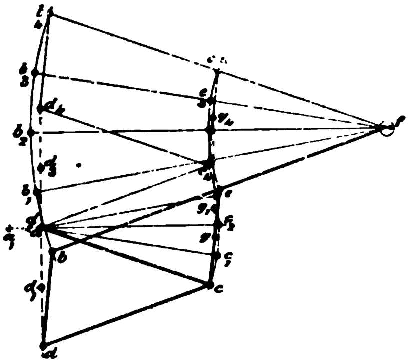

Fig. 67.—Compound parallel motion.

straight line, which fact is utilized in connection with the operation of the air-pump of a beam-engine.

In each of the above cases it often happens to be inconvenient or impossible to place point *a* in line with the path of point *d*, when the length of the radius-rod is increased, and point *a* becomes replaced by point a_1, when the whole of the proportions are affected. For methods of determination of suitable proportions in such cases, for means of estimation of the minimum amount of deviation obtainable, and for descriptions of other arrangements of parallel motion, reference may be made to *Machinery and Millwork*, by Prof. Rankine,

Many questions may, however, be solved by diagram and verified by calculation, based upon the fact that the hypothenuse of a right-angled triangle = the square root of the sum of the squares of the two sides.

The two ends of the radius-rods are fitted with brasses, in general accordance with one of the patterns described for connecting-rods, but of correspondingly reduced dimensions. Similar ends are also used for the main levers at *b* and *d*, Fig. 66. In the adoption of this type the main levers are used for driving the pumps, which calls for increased strength and stiffness, and throws a side strain upon the joints of the brasses at *b* and *d*. For this reason, an arrangement based upon Fig. 67 would give better results. Possibly, however, the best results would be secured by the adoption of case-hardened pins and bushes inserted in bored holes without any means of adjustment. In the parallel motions of steam-engines, the brasses are placed in holes bored in the solid ends of the radius-bars and parallel-bars, and are held and also adjusted by cotters which are secured by pinching-screws.

CHAPTER XXVII.

CRANK-SHAFTS, CRANKS, AND MAIN BEARINGS.

Plain shafts usually adopted.—Crank-shafts for large engines are generally plain turned shafts, slightly varying in diameter in the several parts, and upon which the cranks are primarily secured by shrinking and keying. Bent cranks, or those which are forged with the cylindrical shafts, are confined to engines of small sizes and short strokes.

Material and form.—During the first half of this century shafts were made of cast-iron. Then the invention of the steam-hammer led to the use of wrought-iron for the purpose. The best metal available at the present time is, however, steel of various qualities, from fluid compressed steel downwards. The cylindrical form provides at least as much strength as any other solid section, is cheaply produced, and convenient in application. The bearings must be turned to a circular cross-section, and the remainder of the surface is finished at the same operation.

Hollow shafts.—The strength of a shaft of given diameter is slightly reduced by making it of hollow or tubular section, but its weight is reduced in a much greater ratio. The insertion of a bore-hole also furnishes valuable means whereby possible interior defects

are revealed, and for this reason alone should be certainly adopted in all heavy shafting. The most advantageous proportion for the diameter of the bore is about one-third of the smallest diameter of the outside of the shaft. Less than this diameter of bore is of little advantage. When a larger diameter of bore is adopted, the shaft is of deficient stiffness, and is distorted by the driving of keys, by the application of a heavy flywheel, and by other means.

Cast-iron shafts.—Shafts of cast-iron cannot be excelled for smooth working. Cases have been recorded in which such shafts have been in continuous and satisfactory use throughout a century. Such shafts have doubtless been made of superior material, but the speeds were slow, and the loads were less than would be imposed upon wrought-iron shafts in modern practice. The greatest defect possessed by shafts of cast-iron is their liability to break with absolutely no warning.

Wrought-iron shafts.—Shafts of wrought-iron are about 60 per cent. stronger than the best cast-iron of equal dimensions. They very seldom break suddenly by torsion, though they will do so after exposure to excessive flexure for a long time. A large proportion of wrought-iron shafts which have failed, have done so from the development of cracks, which have originated in a piped centre and extended to the surface. Such shafts contain cavities of irregular shape, which are usually due to the use of a light steam-hammer for forging; and they sometimes fade into imperceptible cracks, which may follow any direction, and which are especially dangerous, on account of their liability to extend. In most cases, a central bore-hole of moderate size will remove the whole imperfect material, and leave the shaft safe to perform heavier work than before

the insertion of the bore-hole. Failures in shafts generally occur in the bearings, on account of their smaller diameter. In such cases the shaft usually loses shape, which causes heating, upon which the shaft is stopped and extensive damage prevented.

Steel shafts. — Steel shafts are not liable to piped centres, as they are not forged under a hammer, but forged by a press from cast ingots. The ingots contain cavities which are closed by the forging pressure, but which fail to seal up solid. The cavities are very much reduced in size if their formation is not entirely prevented by the application of fluid pressure to the ingot before solidification, and the texture of the steel is rendered more solid. This process was introduced by Sir Joseph Whitworth about 1870. In all cases the upper part of the ingot should be rejected, as it contains inferior material and dross. Steel for shafting should usually be made of about 32 tons per square inch tensile strength, with an extension of 32 per cent. in a length of 2 inches. Steel of lower tensile strength is softer and more ductile, and may be used with advantage in work exposed to great and repeated shocks. It is, however, more liable to gall than is harder material. Steel of higher tensile strength is brittle, and liable to fail without warning, breaking usually at one end of a bearing.

Strength of shaft to resist twisting.—A shaft is required to possess strength sufficient to resist the twisting stress due to the force applied to the cranks, and also simultaneously the transverse stress due to the weight of the necessary wheels. A twisting stress tends to divide one part of the shaft from another by a process of shearing along a cross-section. In the normal condition, in which no particle of the material is exposed to a stress approaching the elastic limit, the particles

near to the surface are exposed to greatest stress, which diminishes uniformly to zero at the centre. The intensity of stress at the surface is therefore the chief element to be regarded. This, again, is variable, owing to the variation in the thrust applied by the connecting-rod. It is also different at different parts of the circumference, being greatest at the part next to the crank-pin. In many cases, also, the pressure applied to the crank is modified by the inertia of the reciprocating parts of the engine. In ordinary cases the maximum intensity of stress is from 1·6 to 2·0 times as great as the mean intensity, which is the most convenient unit for application. The mean intensity in tons per square inch prevailing over the surface at any moment =

$$\frac{5 \cdot 1 \times \text{load applied in tons} \times \text{radius of application, in inches}}{(\text{diameter of shaft in inches})^3}.$$

The corresponding diameter of the shaft in inches =

$$\sqrt[3]{\frac{5 \cdot 1 \times \text{load applied in tons} \times \text{radius of application of load, in inches}}{\text{Intensity of stress upon the surface of material, in tons per square inch}}}.$$

If the power is uniformly applied throughout the revolution, as in a water-wheel, the diameter of the shaft required in inches =

$$\sqrt[3]{\frac{\text{Indicated horse-power transmitted} \times 143}{\text{Intensity of stress in tons per sq. in.} \times \text{number of revolutions per minute}}}.$$

But on account of the rectilinear application of the load in a steam-engine, and on the assumption that the pressure upon the crank-pin is uniform throughout the stroke, the required diameter of a crank-shaft in inches =

$$\sqrt[3]{\frac{\text{Indicated horse-power applied to one crank} \times 225}{\text{Intensity of stress in tons per sq. in.} \times \text{number of revolutions per minute}}}.$$

In many cases the work from two or three connecting-rods is applied to separate cranks and taken to one end of the shaft, when the strength of the shaft must be equal to the total amount of work transmitted. The

work will, however, be applied with greater uniformity by three cranks set at angles of 120°, so that a lighter shaft will suffice than would be necessary if the combined work of the three were applied through one connecting-rod to one crank. The condition of loading, in fact, approaches more or less to that of a water-wheel.

Fluctuations in twisting force applied to a crank-shaft.—In all cases a crank-shaft is exposed to stresses which fluctuate. If the pressure of the connecting-rod is applied with absolute uniformity, the twisting force applied to the shaft increases in proportion to the sine of the angle traversed by the crank—subject to small corrections on account of the angularity of the connecting-rod. This increase is from zero at the beginning of the stroke to a maximum at half-stroke, after which the twisting force or moment diminishes in the reverse order, until it again reaches zero at the end of the stroke. The diameter necessary for a shaft thus loaded is obtained by the use of the formula last given, in which the constant adopted is 225. But practically in every case the pressure applied by steam upon the piston is much greater at the beginning of the stroke than at the end, and consequently the shaft must be prepared to withstand a greater stress than the maximum due to uniform pressure upon the connecting-rod. In such cases the diameter is found by the substitution of $\left(225 \frac{\text{Maximum pressure}}{\text{Mean pressure}}\right)$ in the formula.

Intensity of stress to be allowed.—Ordinary shafts of wrought-iron may be loaded up to an intensity of stress of 1·6 to 1·8 tons per square inch, according to the quality of the material and freedom from shock in work. Steel shafts may be loaded to one-fourth more, provided that their transverse and torsional stiffness is not deficient. It is, however, at all times wise to give a little

margin when this can be done, to meet any emergency which may arise, or moderate increase in the future work to be done.

Ultimate torsional strength of shaft.—The approximate load in tons at which a shaft would immediately fail by torsion =

$$\frac{(\text{Diameter in inches})^3 \times \text{shearing strength of material in tons per square inch}}{5{\cdot}1 \times \text{radius in inches at which load is applied}}.$$

When the actual shearing strength is not known, but the tensile strength is known, the latter may be reduced 25 per cent., and adopted instead of the shearing strength.

Strength of shaft to resist bending.—The diameter of a shaft in the several parts must be sufficiently great to impart the strength requisite to resist bending. The maximum intensity of stress due to bending should not exceed 3·2 tons per square inch in iron, or 4·0 tons in steel. The amount of this is quite uniform in a shaft of regular form, and it acts constantly in a vertical direction. But owing to the revolution of the shaft, the stress upon any point in it is constantly changing from tension to compression, and the reverse. If the intensity of stress largely exceeds the above limits, the shaft is sure to break after the lapse of a long or short time, and probably with very little warning. The intensity of stress in tons per square inch =

$$\frac{\text{Load in centre in tons} \times \text{unsupported length in ft., centre to centre of bearing} \times 30{\cdot}6}{(\text{Diameter in centre of shaft in inches})^3}$$

The required diameter =

$$\sqrt[3]{\frac{\text{Load in tons} \times \text{unsupported length in feet} \times 30{\cdot}6}{\text{Intensity of stress in tons per square inch}}}.$$

The strength of a hollow shaft with a bore of one-third of the minimum outside diameter of the shaft is about 1·2 per cent. less than that of a solid shaft of the same

diameter. If the outside diameter of such a shaft is uniform, its weight will be reduced by about 11 per cent. by means of the bore-hole. This reduction in its own weight often makes a hollow shaft rather stronger for supporting external weight than a solid shaft of equal outside diameter.

Deflection of shaft under a transverse load.—The deflection which is caused by the application of weight upon a shaft is often sufficient to call for an increase in diameter beyond that required from a consideration only of the intensity of stress. The deflection in inches of a solid steel shaft, loaded in the centre of its unsupported length, measured from centre to centre of its bearings =

$$\frac{\text{Load in tons} \times (\text{unsupported length in feet})^3}{22 \times (\text{diameter in inches})^4}.$$

That of a hollow shaft will be very slightly greater, and that of a wrought-iron shaft will be 15 to 20 per cent. greater than that of a steel one of equal dimensions. The weight of the shaft itself must be regarded as part of the load; this, being a distributed load, is, in a shaft of uniform diameter, equal to half the weight concentrated in the centre. When heavy cranks or other details overhang upon each end of the shaft, the weight of the shaft may become negligible. In some cases also a considerable pressure, equivalent to weight, is imposed upon a shaft, by reason of the manner in which the power is taken off. It is impossible to give a rule for universal application as to the permissible deflection in heavy shafts, but probably one-fiftieth part of an inch will be found a judicious limit for such a shaft 10 feet long over bearings. Line-shafts of small diameter may be allowed considerably more.

Exceptional cases.—The figures given in connection with the strengths of shafting will meet most cases which arise, and include fair allowance on account of

such shocks and vibration as are inseparable from work. Also for the fact that both torsional and transverse stresses are simultaneously imposed upon a shaft, as in connection with an overhung crank of ordinary proportions, or the twisting and bending stresses due to wheels. In a detailed study of the question, Professor Unwin's *Elements of Machine Design*, Pt. 1, will be found to be of great service.

Division of long span.—In some cases, in which the deflection of a shaft cannot be reduced to moderate limits by other means, the span may be divided into two equal parts, each loaded with one-half of the total weight. When three such bearings are accurately supported, the deflection of each part of the shaft will be only about one-twentieth part of the original deflection. When this necessity arises, the central bearing may be supported from above or below. The former allows the use of tension-rods of small area, and of a minimum width of separation of the two sections of the wheel. Supports placed beneath the bearing are in compression, and when they are of long length the necessary width may become objectionable. If such a bearing be carried from above, it will be usually found necessary for safe working that the construction shall be quite separate from any floor, especially if this is wholly or partially used for warehouse purposes, the object of which limitation is to avoid the variations in level due to variations in load and in temperature. If, however, this cannot be avoided, the weight should be carried upon laminated or locomotive springs, and adjustments made according to measurements taken from rim to rim of the two sections of the wheel, in eight places, and repeated after turning the wheel through half a revolution.

Canting of bearings on account of deflection of shaft.—The deflection of a heavily-loaded shaft, supported in

two bearings, is such that each of the bearings must be set to suit, or the shaft will not work properly. The angle of such departure from a level condition may be approximately obtained by measuring the total deflection of the shaft as the rise, and one-fourth of the distance from centre to centre of the bearings as the length of the slope. This is, however, subject to slight correction on account of the differences in elasticity of material, and of the proportions adopted in reducing from the larger diameter at the centre of the shaft to the smaller diameters at the bearings; by a more detailed calculation the latter element of uncertainty can be eliminated, but not the former. Messrs. Hick curve the beds of the main-bearing blocks, so as to allow self-adjustment of bearings.

Deflection at crank.—Deflection to an extent much smaller than that last described also occurs at the end of the shaft, in the plane of the piston-rod, caused by the alternate thrust and pull of the connecting-rod. This, however, seldom reaches an important amount, but it may be approximately calculated according to the rule above given, substituting twice the force of the connecting-rod for the load, and twice the distance from the centre of the connecting-rod to the centre of the main bearing for the length.

Changes in diameter.—Changes in the diameter of a shaft should be made as small as possible. As a rule, a change should be made by a taper part of as great length as practicable, leaving only such parallel surface as is required for receiving wheels or eccentric sheaves. All unavoidable shoulders should be well rounded in the corners. Heavy bored wheels, cranks, or other details should have the corners of the bore eased off. All these measures tend to prevent the occurrence of intense local stress, which, being often repeated, is apt

to produce minute cracks, which are sure to extend and ultimately cause the sudden rupture of the shaft, however long it may be delayed.

Freedom for longitudinal movement.—A crank-shaft should be allowed slight end play or free longitudinal movement. One-sixteenth of an inch will usually suffice, and will exercise a powerful influence in maintaining smooth bearing surfaces. For this and for reasons connected with the strength of the shaft, such a shaft should be free from the imposition of end thrust. If a continuation of it is required to carry bevel-wheels or equivalent details, special collar or thrust bearings should be provided. If these are at a distance of 15 to 18 feet from crank-shaft bearings, the expansion and contraction of the shaft by reason of temperature will ensure such variation as will be of material assistance in maintaining the bearings in good condition. A rather less amount of end play is of importance in a crank-pin.

Area of bearing surface.—The diameter of a shaft must be sufficient to give adequate area of bearing surface, as explained in connection with main bearings. If, however, the shaft is sufficiently stiff, and space is available, the requisite area of bearing surface may be obtained by an increase of length. A strong impression has prevailed to the effect that an increase in diameter is accompanied by a greater risk of heated bearings, on account of the proportionate increase in surface speed. It is very difficult, if not impossible, to find evidence of the existence of such a condition at any time, and certainly not in connection with the adoption of copious lubrication, as in present good practice.

Block, or web, and disc-cranks.—The cranks of large engines are almost invariably made of wrought-iron or soft steel. Disc-cranks of cast-iron are occasionally adopted, with the object of balancing the crank-pin

and the connecting-rod end. Disc-cranks are, however, difficult to design and cast quite free from original strain, and are consequently liable to break either spontaneously or in consequence of accident. Cast-iron block-cranks, like cast-iron shafts and connecting-rods, are now rarely used in new work. The outline of wrought cranks is usually as shown in Fig. 68, in which the width of the web is reduced below the diameter of the bosses. This presents a good appearance, but is likely to entrap the unwary attendant, who, notwithstanding all regulations to the contrary, places his hand in danger "only once." For this reason the outline shown in full

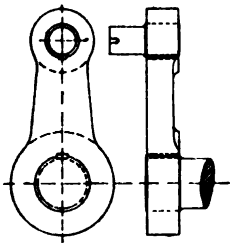

Fig. 68.—Block-crank, ordinary pattern.

lines in Fig. 69 is distinctly superior, while it presents greater stiffness than the first. The appearance of a crank is considered to be improved by a projecting boss or facing surrounding the shaft, and by a slight projection of the shaft itself. But these survivals from the cast-iron era are also found to be practically dangerous, and should be avoided. The length measured through the shaft bore-hole is often made greater than that through the crank-pin bore-hole; this is done under the impression that the proportion of diameters should be regarded, if not closely followed. There is, however, no good reason for this, and experience shows that more cranks work loose on the crank-pin than on the shaft. The

large end of a crank-web should always be continued as shown in dotted lines in Fig. 69, for the purpose of providing a balance for the crank-pin, and about one-third of the length of the connecting-rod.

Fitting and shrinking cranks.—Cranks are generally bored for the shaft and the pin to a shrinking fit, then heated, cleaned, and put together. The shaft should be prepared for sunk keys in either one or two places, corresponding key-beds to be cut in the crank, and keys well fitted. When a crank is shrunk on a shaft or

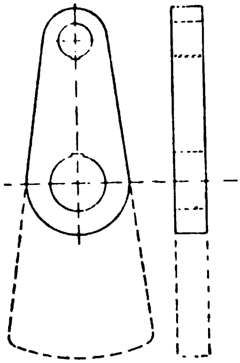

Fig. 69.—Block-crank, stout pattern.

crank-pin of uniform diameter, the inner end of the bore should be eased all round to prevent cutting the shaft or crank-pin, and thereby commencing a crack. This need not reach more than one or two-eighths of an inch into the bore, and at the face should be only just appreciable in amount. Shafts and crank-pins are occasionally broken near to the face of the crank, in a manner showing that the sharp *arris* has promoted if not caused the accident. In some cases the breakage is about half-an-inch inside the crank, and in others it varies, some parts being nearly flush and others within;

in all such cases, however, the breakage occurs with little or no warning, and is short and nearly square. In many cases, shafts or crank-pins are made of one uniform diameter in the crank and in the bearing, the necessary collar being formed by the face of the crank. This gives more uniform elasticity, and is perfectly efficient if the work is accurate, so that the crank faces work truly.

Accuracy.—The holes for shaft and pin should be accurately bored at right angles to the face of the crank, for the sake of true appearance in running. But a consideration of the greatest practical importance is that the two should be absolutely parallel to each other. Many ways whereby this may be checked will occur to an expert erector. The most simple and absolutely conclusive test can, however, only be made after the engine is put together. This is done by removing the cross-head pin, or the strap surrounding a solid pin, and closing the crank-pin brasses upon the crank-pin. The shaft is then turned into the four quarter positions, and the correspondence of the cross-head and the end of the connecting-rod observed. If the latter falls a little to one side in one position, and to the other side when the crank is in the opposite position, the shaft or the crank is in fault; either the holes are out of parallel, or the shaft or the crank-pin is bent, or the shaft is not lined square with the cylinder and cross-head slides. If the connecting-rod falls central in every position, the whole is in accurate adjustment, and the crank is truly bored. If the connecting-rod falls towards the same side of the cross-head by an equal amount in all positions, the crank-shaft, crank, and crank-pin are all true, but the crank-shaft requires to be moved in a longitudinal direction or the connecting-rod is bent. The cross-head brasses may be then re-coupled, the crank-pin brasses

taken apart, and tests made in the same positions, the results of which will furnish additional evidence as to whether the shaft is truly square with the line of the cylinder and the cross-head slides. In making this test care should be taken to keep the shaft in the same longitudinal position in the bearings throughout.

Crank-pins.—Crank-pins, like shafts, should be arranged of diameters as uniform as possible, the bossing to enter the crank being little more than sufficient to allow the cutting of a key-way. If a collar of larger diaméter is required behind the connecting-rod brasses, a loose ring may be provided. A loose washer is often provided at the outer end, fixed by a set-screw; when continuous lubrication is applied from the front, the washer may be fixed by three screws to avoid the centre. The length of the crank-pin bearing should allow a little end clearance, as already described.

Multiple cranks.—Shafts are often adopted in which the power from one or two cylinders is transmitted through one crank, in addition to that from its own cylinder, such a condition occurring when the power from two or three parallel cylinders is all sent in one direction. Crank-shafts of this class for engines of moderate and large dimensions are built from separate shafts and crank-webs. In this process care is required to ensure that the several cranks lead in the proper order, and that the angles are practically correct. But the most absolute accuracy is required in boring the crank-webs, not only with respect to parallelism, but as to distance apart of centres of holes. The cranks must also be shrunk upon the pins in such a manner that the several lengths of shaft will be quite true with each other. In the latter respect a satisfactory degree of accuracy can only be secured by arranging for the support of the several lengths of shaft upon

blocks placed accurately in line before the crank-pins are placed in position for shrinking. The whole of the appliances must be kept scrupulously clean, or good work cannot be obtained. The truth of the work may be afterwards tested by turning in its bearings, or by lifting the whole from its bearings in the engine, and supporting by light stiff V blocks of wood laid upon the frame. When the shaft is turned round in such a position, each block should support a constant pressure. But if any part of the whole is out of truth, either in consequence of defective shrinking or of inaccuracy in the boring of the crank-webs, one or other of the blocks —especially against a crank-web—will be relieved of weight when the shaft is in one position, and another block when the shaft is turned round into another position. If such defect appears to attain its maximum when the adjoining crank is in a vertical position, the inaccuracy is caused by incorrect boring. If the defect is shown to be greatest when the adjoining crank is in a horizontal position, it arises from incorrect shrinking of the cranks upon the crank-pins. Small strips of tin plate or zinc, well flattened and placed in clearly-defined positions, may be used beneath the blocks as feelers in testing the pressure supported by each.

Relative liability to slip.—If a built crank-shaft is truly accurate throughout, and is supported in equally accurate bearings, close to each side of each crank, of each of which bearings both the lower brass and the cap are quite rigid, each crank-pin will possess sufficient strength if made slightly larger than the ordinary pin suitable for an overhanging crank. But in the best engines the bearings fail to answer to the condition of perfect rigidity, while the majority of engines are distinctly defective in this respect. As a result, cranks of this class are found to be about equally liable to slip on

the pin or on the shaft. The two bored holes are therefore made nearly or absolutely equal in diameter, the working surface of the crank-pin is of diameter equal to that of the main shaft, and the keys are of equal strength, and similarly disposed in shaft and crank-pin. The body of each crank-web should therefore be made parallel, and each end finished to an equal semi-circle, except as may be required to be modified on account of balancing. If a crank should slip appreciably upon the shaft, it may be re-keyed to prevent further movement, and put to work again. But if it should slip on the crank-pin in the most minute degree, its working accuracy is destroyed for the time, and can only be restored by heating the web to loosen the crank-pin and re-shrinking, or by other equivalent means.

Fixing and pressing.—By the exercise of great care, accuracy can be attained by shrinking or pressing a pair of crank-webs on the crank-pin, and *afterwards* finishing the bore for each shaft parallel to the crank-pin, of which the centres should be left for convenience in adjustment. If this is accurately performed throughout, uniformity in distance of centres in the two parts and parallelism are secured in one operation. If, however, the cranks are afterwards heated for shrinking upon the shafts, the crank-pin ends must be kept cool or they will slip. It is therefore better to secure the cranks on the shafts by pressing and adjusting the diametrical fit so that the force applied in a longitudinal direction shall be about ·4 ton per square inch of surface in contact, the surfaces being very slightly greased before putting together to prevent galling.

Crank-pin solid with webs.—Some engineers make one large solid forging to form one crank-pin and two webs, as in Fig. 70. This is a good plan if good proportions are adopted, but there is a great temptation to make

the webs too light in the parts near to the crank-pins. Great care must be taken in boring the shaft-holes in the webs, to ensure accurate parallelism with respect to the crank-pin. Dickinson's built crank-shafts are of this class, but each length of shaft is terminated by a flange, exactly as for coupling to another by flange bolts, of which the proportionate strength may be obtained, as in the case of plain shafting. The web should fit accurately round the edge of the flange, but not tightly shrunk, as that would interfere with the

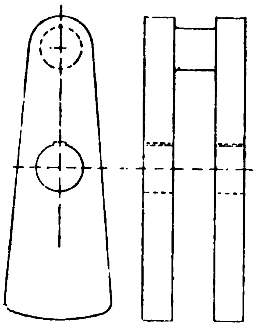

Fig. 70.—Double-webbed crank to receive plain shafts.

facility for taking apart and replacing in case of damage.

Truth and condition of surface.—Accuracy of form is of great importance to a shaft. When revolving in its bearings, the shaft floats on a film of oil of very small thickness. If the complete shaft should be out of truth, or if its cross-section departs from a true circle by even a fraction of the thickness of the film of oil, the shaft in its revolution will act the part of a pump, and rapidly work the oil out of the bearing. No direct mischief is likely to arise from a moderate action of this kind, if the supply of oil is well maintained. But if

the supply should be interrupted for a short time, the bearing becomes dry, and heating is caused. In all cases such irregularity gives rise to increased resistance, and the cause should be avoided. A smooth surface is also advantageous in minimizing the resistance to motion. This can only be secured by a fine clean finishing cut, and subsequent polishing with emery graded finer and finer, followed by ground turkey stone or equivalent material. No part of such bearings should show any tendency to seize or hold fibres in cotton waste when rubbed across. Great care should also be taken to prevent bearings from receiving indentations, abrasions, or damage of any kind after finishing.

Conditions to be observed in main bearings.—Main bearings are generally loaded to a great pressure, and therefore require to be constructed with great strength and rigidity. The shaft should be perfectly free to revolve with the least possible resistance from friction, but must be restrained from transverse or longitudinal motion of any kind, except the slight end play already referred to. The appliances used for supporting the shaft should be free from excessive liability to distortion and wear, and should possess means of adjustment to compensate for wear.

Flat and angular pedestals.—For supporting the crank-shafts of beam-engines, flat pedestals of the ordinary type, but especially strong and well fitted, were used. The use of these was also continued in horizontal engines, but without any good reason was abandoned at an early period in favour of angular pedestals, precisely similar, except that the cap and the parting of the brasses are turned over at an angle of 40° to 50° with the horizontal plane. The fact that some engineers turned the cap towards the cylinders while the majority took the opposite course, supports

the opinion that the flat or level-jointed pedestal is quite as suitable as the angular form. Angular pedestals are often arranged so that the oil is scraped off the journal at the lower side of the joint before it reaches the intended point of application at the bottom of the bearing. In most engines the half weight of the fly-wheel, shaft, cranks, &c. which rests on each bearing is quite sufficient to prevent the shaft from pressing against any part of the upper brass of an angular pedestal until the lower brass is very much worn. This view is confirmed by the fact that brasses which are worn so far as to require renewal have suffered most in the part directly under the centre of the shaft, any deviation which may exist being towards the lower part of the joint of the brasses; also by the fact that in case of heated bearings there is seldom any hesitation felt in running the engine for a time without upper brass. Angular pedestals are, however, often valuable for second-motion shafts, and other situations for resisting lifting.

Arrangement of brasses.—Plain flat or angular pedestals are bored, and the brasses are turned on the exterior to fit the pedestal. Snugs are generally provided on the brasses to prevent their rotation, which snugs interfere with the turning of the brasses in the lathe over some part of their length. This portion of the brasses is cast a little thinner than the rest and left rough. The best position for the holding snugs is close to the joint of the brasses. In such position they are convenient for securing the brasses together for turning and boring. After these operations are finished, the snug is cut from the non-effective side of the lower brass, so that it can be easily turned over to the upper side of the shaft for examination at any time when the weight of the shaft is relieved.

Uniformity of thickness in brasses.—Thin brasses are much more subject to distortion than thick ones. Those on flat or octagonal beds, or otherwise irregularly formed, are also liable to distortion and to break across the thinner parts. In some arrangements it is not convenient to provide brasses of uniform thickness throughout,* when the minimum thickness should be somewhat in excess of the thickness necessary, in case uniformity is possible.

Special security of holding brasses.—An excellent arrangement of main bearing is shown in Fig. 71, which Mr. Seaton has found "most successful in vertical

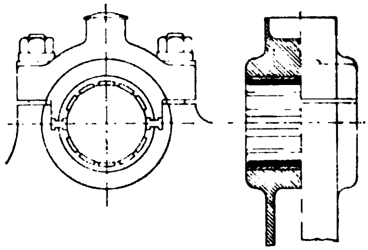

Fig. 71.—Main bearing for vertical engines.

engines of all sizes" (*Manual of Marine Engineering*). Two H-shaped strips are adopted to prevent the brass from closing upon the shaft, as explained in the chapter on maintenance of machinery. By this means the entire fitting is maintained firm, no snugs are required, the whole of the outside of the brasses may be finished in the lathe, and the lower brass is free to remove for examination upon a very slight relief of the weight of the crank-shaft. But no means of side adjustment for wear are provided. In vertical engines, to which this type of bearing is chiefly applicable, the bed-plate is generally a compact casting, either in one piece or in several pieces, securely bolted together. This can be

bored out on a boring-table to receive brasses for the requisite number of bearings, all with perfect and permanent accuracy within the limits of deflection of the bed-plate.

Main bearing with side-blocks.—When the force exerted by the connecting-rod of a horizontal engine exceeds one-third of the weight upon the crank-bearing, side-blocks to the bearing should be provided, as in Fig. 72, with means for very accurate adjustment, so

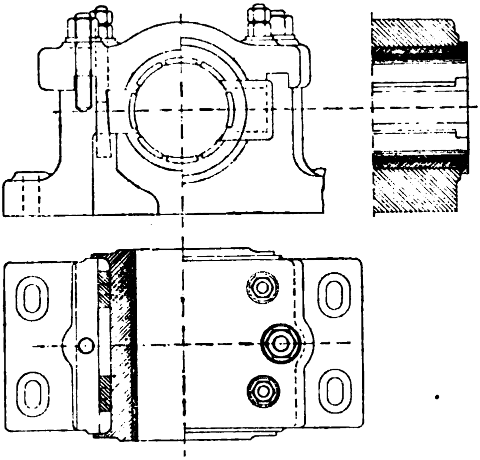

Fig. 72.—Main bearing, with side adjustment.

that the pressure may be kept constantly in the bottom of the brass, and injurious percussion and chafing prevented. Wedges are usually adopted for fine adjustment, as in the Allen engine, being set up by vertical screws, so that any required degree of nicety can be attained. In large bearings, either two or three wedges are used on each side, which are disposed to avoid the cap-bolts as much as possible, or one large wedge-piece on each side is drawn up by two or three scerws.

Horizontal wedges are also adopted, drawn together by screws, and made to such an angle that one or other will yield a little if required, the whole giving the means for most delicate adjustment. In inferior work the side-blocks are adjusted by set-screws pointing towards the shaft, and locked by nuts. In many bearings with wedge adjustment, the bed upon which rests the bottom brass is planed flat, and a bottom brass of considerable weight and strength is provided. In others, as the one illustrated, the block and cap are bored to receive the bottom and top brasses. The dimensions of the details must be such that the length multiplied by chord of bearing surface upon bottom block gives a bearing area sufficient for the weight imposed. In like manner the area (length × vertical chord) of the side-blocks must suffice to withstand the side-thrust of the connecting-rod. The cap is comparatively unimportant, its chief duty arising in case of accident. A troublesome bearing is sometimes run without cap, showing that the cap usually performs little duty. The cap should, however, never be left off for one moment longer than is necessary during work. In all large horizontal engines of high class, bearings with side adjustment are almost invariably employed at the present time. On the whole this is wise practice, though they are not in every case absolutely necessary. By leaving a small amount of clearance on each side of the bottom brass when fitted on a planed bed, sufficient means are provided for the necessary adjustment of the shaft after the fixing of the bed-plates, provided this is done with reasonable care. In the use of ordinary pedestals, this adjustment is provided by moving the whole block, and keying against snugs by means of fitted keys. In vertical engines there is, as a rule, very little need for side adjust-

ment of crank-shaft, unless on account of the way in which the power is taken off. When this is done in such a manner as to impose transverse stress, adjustable side-blocks may be provided as in a horizontal engine. In Fig. 72 the cap is shown provided with fitted lugs at each end, with a view to distribute any horizontal pressure over the two vertical members of the pedestal-block.

Bearing materials and pressures.—Gun-metal of good quality is most extensively used for bearing-brasses, and is perfectly satisfactory if well designed and fitted, and if not overloaded. With drop lubrication, the total pressure upon the bearing should not exceed 400 pounds upon each square inch in the longitudinal section of the bearing, or the length multiplied by the chord, or by the diameter in case the bearing surface extends over the entire half circumference. It is, however, better to keep the pressure well below this rate. With copious and reliable lubrication, 600 to 700 pounds per square inch may be taken as a maximum. Phosphor bronze will carry a greater load than gun-metal, but if a great excess in the load is adopted, and by any chance the lubrication should fail, the shaft is sure to suffer most severely. White metal of various compositions is used with success, either in the form of large surfaces or of strips driven into recesses cut into the brass in a longitudinal direction. Strips thus fitted may be easily knocked out and renewed when worn. The whole of the pressure should be sustained by the white metal, the brass or iron matrix which supports it being cut back to give a good clearance from the bearing. The white-metal strips should be arranged from 2 to 4 inches in width, and the pressure should not exceed 250 pounds per square inch in strips 2 inches wide, or 500 pounds in strips 4 inches wide.

In crank-pin bearings the mean pressure may be one-half greater. In this connection the actual width of strip is to be measured for classification as to admissible load per square inch, but the load allowed to each strip should be estimated upon the difference between the dimensions *a* and *b* in Fig. 73, multiplied by the length of bearing in inches. White metal should not be used for cross-heads or other reciprocating bearings. Cast-iron is an excellent material for bearings, if carefully

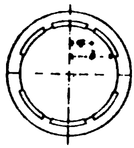

Fig. 73.—Main bearing, section of brasses fitted with white metal.

fitted and attended to in the first instance. Oil should be most liberally applied to a cast-iron bearing for a short time, when the cast-iron will acquire an excellent glazed surface. Though the same degree of attention is not again required, it cannot endure great neglect. The permissible pressure is from 50 pounds per square inch in shafts of small diameter to 1000 pounds in slow-running, well-lubricated bearings of chilled cast-iron accurately ground.

CHAPTER XXVIII.

ECCENTRICS AND ECCENTRIC-GEAR.

Eccentric motion.—In rare instances, when one end of a crank-shaft is available for the purpose, a small crank is fitted upon it to actuate the slide-valves. In most cases, however, it is necessary to obtain the same movement from a point in the shaft situated at some distance from the end, when an eccentric is adopted. This is equivalent to a crank to give the same throw, or whose centres are at the same distance apart, and of which the second centre may be in any position within or without the shaft.

Work of eccentric.—The power required to move a slide-valve may be estimated on the assumption that the co-efficient of friction may at times reach 0·15. At such times, the force required to move the valves of a cylinder with short ports and separate plain slides at each end amounts to several tons.

Fixing and proportions of eccentric.—The eccentric-disc or sheave is generally secured to the crank-shaft by a hollow-backed key for facility of adjustment. In many cases a flat might be cut upon the shaft, and the position of the eccentric permanently secured, with advantage. This is, however, seldom done, and the strength of the boss requires to be made sufficient to

withstand the driving of the key with sufficient force to prevent slipping. A second key is sometimes added, at 90° from the first one. The sheave is generally made in halves, for convenience in taking apart and replacing. The two halves are made of cast-iron, jointed by planing and fitting, then secured by tight-fitting, turned cotter-bolts, the recesses above the heads being stopped by cast-iron plugs, which are turned off with the rest of the surface. It is very seldom that a sheave requires to be removed after the engine has been put to work, but when made in halves it may be placed in position at the latest moment, so that up to that time it may be kept out of the way. A solid sheave is, however, neater and rather better, and in many cases may be freely allowed. The width of boss is decided chiefly with reference to the strength for keying, and is maintained almost undiminished to the edge of the sheave. A narrow flange should be provided on each side, to prevent the clips from moving sideways from position. The width and area remaining for bearing surface are usually more than sufficient to bear the pressure, which should not exceed 60 pounds per square inch, calculated upon the width and diameter of bearing surface. In many cases the clips are restrained from side movement by means of a narrow central tongue formed upon the sheave, fitting a groove in the clips. By this means the entire width of the sheave is available as surface for bearing, but the value of the whole is impaired by its divided condition to such an extent that it is usually worth less than the smaller area in one piece. Eccentric clips are made in halves, to allow passing over the sheaves. Usually the two halves are planed and fitted together before boring. In other cases the two clips are cast together, bored, and afterwards cut apart, packing-pieces being fitted in the thickness of the

parting cut. The packing-pieces are made of iron or brass, and are found convenient to reduce in closing the clips together after use.

Materials used for eccentric.—The clips are usually made of cast-iron, and they wear very well without special preparation. They are, however, often bushed with brass, which increases the cost without securing any benefit. In some cases they are made of cast-steel, which wears to an excellent and almost everlasting surface, but leads to excessive wear of the cast-iron sheaves. A much better plan is to make the sheave of cast-steel, and the clips of cast-iron. The fit will not then be destroyed by the wear of the latter as it is by the wear of the sheaves. Brass clips were formerly used, but are now abandoned. They possessed no advantage, and the temptation to save cost in the metal constantly led to weak proportions and breakage. Even with the most ample proportions adopted, brass clips always possessed an objectionable degree of flexibility in work. In all cases, the thickness of the outer clip should be made slightly greater at the crown than near to the bolt-bosses.

Eccentric clip-bolts.—The two bolts which connect the clips should be placed at the smallest possible distance apart, so as to reduce the bending moment upon the clips to its lowest amount. Under such conditions the length of the bolts will be increased, but this is no disadvantage. The clip-bolts should be provided with lock-nuts.

Attachment of eccentric-rod to eccentric-clip.—In the oldest efficient plan, the eccentric-rod is secured to the inner clip by means of a dove-tail recess in one side of a projecting tail, into which the rod is carefully fitted and secured by several fitted bolts, which run parallel to the shaft. This is an excellent but costly

plan. The rod is thrown a little towards one side of the centre line of the sheave, but this is often required to be done to meet the position of the valve-spindle, and if not so required the eccentric may be returned by cranking. The eccentric-rod may be secured to a steel clip—but not in good practice to a cast-iron clip—by cottering against a shoulder. The best arrangement is that shown in Fig. 74, in which the eccentric-rod is

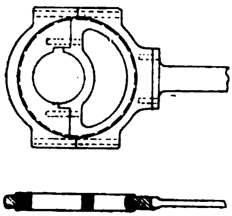

Fig. 74.—Eccentric.

terminated by a flat table or palm, which is fitted to the clip by a planed joint and secured by two studs or set-screws tapped into the cast-iron. In this arrangement the eccentric-rod naturally follows the centre line of the sheave, but it may be conveniently placed a little towards one side if required.

Sectional form of eccentric-rod.—The eccentric-rod for a horizontal engine is usually either rectangular or approximately so in section. The thickness of this detail is very small in comparison with its length, and much vibration exists when the stress per square inch exceeds 600 pounds upon the smallest section; it is, however, better that it should not exceed 500 pounds. The smallest part of the rod should be farthest from the eccentric. A good general form for rectangular rod is shown in Fig. 75, and an alternative one for polished rods of smaller dimensions is shown

in Fig. 76. A rod of tubular form would be better as to absence of vibration due to compression. Depth is, however, required to impart vertical stiffness to the rod. In vertical engines this reason does not apply, but in all engines a certain amount of stiffness in the eccentric-rod is useful to prevent the clips carrying round, if allowed to seize upon the sheave by reason of neglect.

Fig. 75.—Section of large eccentric-rod.

Fig. 76.—Section of small eccentric-rod.

Gear actuated by eccentric-rod.—The end of the eccentric-rod next to the cylinder is often made as a plain strap end, being a reduced copy of such a one as is applied to connecting-rods and shown in Fig. 63. This may be either single or forked. When slide-valves are used, and the line of the eccentric-rod can be made to coincide with that of the valve-spindle without or with a small amount of cranking, the connection can be made directly. When, however, the two lines cannot be brought to coincide, the best connection is made by means of a short stout cast-iron lever, carried by a rock shaft, and with a stud through the top boss, one end of such stud engaging the eccentric-rod, and the other end the link which is attached to the valve-spindle. In some cases, two separate wrought-iron levers are employed, keyed separately upon a shaft; but the torsion of the shaft and the deflection of the levers is apt to cause great unsteadiness of motion. In either case a stout shaft of moderate length is essential to success. In connection with an eccentric-rod of great length, a joint and

supporting lever should be provided near to the centre of the length. When a rocking-lever is required to connect the centre lines, its position may be decided with reference to the same object. A good joint for eccentric-rods is shown in Fig. 77. The end of the eye, where it bears against the brass die, should be strictly concentric with the pin, or the die will be alternately tight and slack in work. The proportions of bearing surfaces in all valve-gears should be in accordance with the conditions affecting cross-head pin surfaces. A valve-spindle is usually supported by a bracket outside the stuffing-box, leaving good space for access to pack the

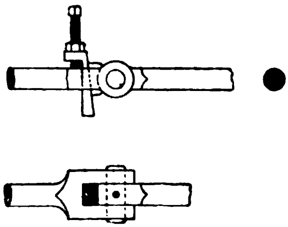

Fig. 77.—Joint for eccentric-rod.

box. An excellent arrangement of bracket is of cast-iron, bored to fit a bush of cast-iron which fits the valve-spindle, and is easily renewed when worn. A cross-head is often provided upon the valve-spindle, in such a manner as to allow an adjustment by means of a screw, accessible without removing a valve-chest cover.

Light gear for light work.—Eccentrics and rods for driving trip-slides or wrist-plates for Corliss valves are arranged in the same way as those for driving plain slide-valves, but are exposed to much lighter work, and may be made of correspondingly smaller dimensions.

Non-interference with access to details.—The provi-

sion of valve-levers (as well as all other details) should be so arranged as to allow access to the piston of each cylinder without interfering with any other details. Generally, the hindmost or uppermost cover of each cylinder is the one to be removed to give access to the junk-ring for examination of pistons, but this is not always the case.

CHAPTER XXIX.

ENGINE-FRAMING.

Duty of framing.—Framing is required to maintain the several parts of an engine in accurately adjusted relation to each other, notwithstanding the imposition of uniform or variable stresses due to the work performed, or arising from defects in the supporting power of the foundations.

Bed-plate for vertical inverted engine.—In a vertical inverted engine the crank-shaft is supported in bearings constructed in a bed-plate. The bed-plate may be cast whole, or in sections, well secured together by planed and bolted joints. The former is the best arrangement when possible, but the latter is often necessary on account of difficulties in connection with casting and transit. The strength of the frame or bed-plate must be sufficient to maintain the shaft in true line, under the alternating stresses imposed by the several connecting-rods and the conditions of support. Each bearing part is required to act as a beam in opposition to the lifting force applied by the adjoining connecting-rod or rods. Usually also as a beam in opposition to the downward pressure of the rod or rods; but this may be largely reduced by carefully-fitted support exactly

beneath the bearing. A bed-plate of ordinary design is made in the different parts, with two vertical sides and a horizontal top plate connecting the two. The material is thus better adapted for resisting an upward force applied at the centre of the length than a corresponding downward force, so that support beneath the bearings is a necessity for a weak bed-plate, and an advantage in all cases. A pounding action in work is often caused by slackness or inaccuracy in the crank-shaft or crank-pins. For this reason, if the strength of the cross-bars is decided by calculation based upon the statical load applied, the intensity of stress upon cast-iron should not be allowed to exceed 1·5 tons per square inch, and upon cast-steel one-fourth more.

Standards for vertical inverted engine.—In the simplest form of the vertical inverted engine each cylinder is supported upon standards, erected upon the bed-plate and devoted to no other purpose, or only to the support of the cross-head slides in addition. During the up-stroke, the connecting-rod is in tension, and the standards in compression. During the down-stroke, the opposite conditions prevail. The length of the standards is therefore greater during the down-stroke than during the up-stroke. Cast-iron standards, loaded at the rate of one ton per square inch in sectional area, increase or decrease in length by about one seven-thousandth part, so that such standard 12 feet high will be about one twenty-fourth of an inch greater in length during the down-stroke than during the up-stroke. When this occurs with uniformity, no serious objection need be advanced, but if the area of one standard largely exceeds that of another, or if they are differently disposed, so as to suffer unequal loading, the truth of line of piston-rod becomes seriously affected, and vibration is caused. Wrought-iron extends and contracts

under load only to about half the extent which cast-iron does. Therefore standards of wrought-iron may appropriately be loaded twice as heavily as those of cast-iron. In all cases it is unwise to apply heavy loads to the standards when it can be avoided, as it is then very much more difficult to entirely prevent vibration. In all cases the standards should be arranged so that equal loads are applied to each point in the section of each standard. The leading condition to be observed with a view to the latter object is that the centres of the front and back standards shall coincide with the central plane through the piston-rod, thereby avoiding the bending action which arises in any other case, and of which a striking instance is referred to in Chapter XX. In the simplest case in which standards are applied to observe this condition, the front standards are immediately in front of the connecting-rods, and the back standards immediately behind. The former interferes very much with sight and access to the rod, and both are attached to the bed-plate at a considerable distance from the main bearings, so that the bed-plate is exposed to great bending action. Both these objectionable features are relieved in large high-class marine engines by dividing the lower end of the front standards—if not the back ones as well—and spreading the two portions well apart, where they are attached to the bed-plate. In some engines, with three parallel cylinders and separate cranks, the centre standards are omitted, so that the whole of the stress due to the centre cylinder is equally distributed over the four standards of the other cylinders. The arrangement is, however, not perfect, as the joints connecting the several cylinders are subjected to much pressure, and sometimes to vibration in combination. In many cases the front standards are omitted,

for the sake of better access to the crank-shaft, and turned vertical stays of wrought-iron are substituted. If these are made of sectional area less than one-half that of the cast-iron standard, some sacrifice in stability will be incurred, unless they are placed at a correspondingly greater distance in front of the plane of the crank-shaft than the back standards are behind the same plane.

Variation in length of standards under stress.—The amount of stress imposed upon each standard at successive periods by reason of the thrust of the piston-rods may be easily calculated. The exact amount of change in length upon standards of varying section is rather more tedious but not difficult to calculate, from which the twisting action may be deduced. The amount of this varies approximately with the height, and inversely according to the distance apart of the two standards.

Combination of condenser with standard.—If a vertical engine is provided with a surface condenser, this is usually placed upon, or abutting against, the back part of the bed-plate, with the air-pump and circulating-pump behind it. An upward continuation or arm of the condenser—similar to the upper part of a front standard—is provided to carry the back part of each cylinder, the front part of each being carried by a standard, as already described. The stiffness of the condenser causes generally a great reduction in the visible movement of the cylinders arising from difference of extension in standards, but the twisting effect upon the cylinders is not extinguished.

Pressure from cross-head.—In addition to the vertical tension and pressure applied to the standards by means of the piston-rod and connecting-rod, a transverse or horizontal pressure is applied by means of the cross-head, due to the angularity of the connecting-rod. So

far as this is due to direct pressure of steam, it always acts in the same direction, so long as the motion of revolution continues in the same direction. The amount, however, fluctuates from nothing at each end of the stroke to a maximum near the centre of the stroke. A slight variation is in some cases caused by the adoption of compression of steam in the cylinders. The transverse stiffness of the standards, with or without assistance from a surface condenser, where one is provided, must be sufficient to withstand this fluctuating pressure from the cross-head, or bracing must be resorted to, or vibration will result.

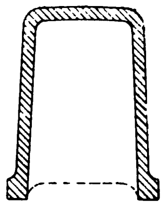

Fig. 78.—Section of framing for horizontal engine.

Framing for horizontal engines, original type.—The original system of framing for horizontal engines was developed from timber construction, in which longitudinal beams are connected by cross-beams or bars tenoned into the longitudinals and secured by bolts. In the best examples of this type the wooden construction is replaced by an iron casting of hollow section, deeper than ordinary whole timber beams. The width across the section is greater at the bottom than the top, and the upper surface is slightly convex, with the angles well rounded off. Fig. 78 shows the form of such framing in cross-section. Rigidity is ensured by depth and thickness of metal. The ledges along the bottom serve the double purpose of increasing strength

and increasing the bearing surface upon the foundation. In a well-proportioned bed-plate the strength of the upper portion exceeds that of the lower portion, on account of its receiving a greater amount of stress, due to the steam-pressure applied along the piston-rod in a plane above the bed-plate. The two sides are connected by cross-ribs, dotted in Fig. 78, and the entire section is generally duplicated under each cylinder and under the slide-bars; and is increased in width between the slide-bars and the crank-shaft bearings. At each point where cylinders, slide-bars, main-bearing pedestals, or other fittings are attached, the top surface is planed, and a hand-hole provided for access to the under side of the bolt-holes. The framing is cut away where required, for the passage of exhaust-pipes, for clearing the connecting-rod, or for other purposes. In each case the necessary strength of the whole should be scrupulously maintained. Hand holes should be placed as nearly as possible upon the neutral axis of the section, with respect to transverse strength. Such a bed-plate is cast in one piece up to a length of 30 feet, beyond which it is made in two or more pieces carefully fitted together and secured by dowelling. In all such cases slipping of the joint should be absolutely prevented by steady pins, keys traversing the joint, or the use of turned bolts well driven into rimered holes.

Framing for horizontal engine, direct type.—The system of framing for horizontal engines in which the working stresses are directly opposed is rapidly extending in favour. In its complete form, as applied to large engines, the framing, properly so called, only extends forward from the foremost cylinder. The after part of the bed which carries the slides terminates in a plate, which forms the fore cylinder cover. The upper part of this plate is bracketed to the bed-plate in such a

manner as to give a strong and rigid connection for resisting the longitudinal stresses which arise alternately in opposite directions during work. The after end of the cylinder then only requires to be supported as to weight, which must be done with special care. By this means the space beneath the cylinder is left free, so as to facilitate the application of valves of the Corliss or other type below the cylinder. In most cases valves in this situation are exclusively exhaust-valves, the steam-valves being applied above the cylinder. When framing of this type is adopted for tandem engines, the two cylinders in line are usually connected by distance-pieces. In some cases these are of cast-iron, turned at each end to secure truth of line in the cylinders. In some cases short sections of bed-plate are used, well provided with flanges bracketed to some distance above the plane of the piston-rod. In other cases two or three strong wrought-iron bolts are used, screwed and double-nutted or shouldered at one or both ends, fitted to bosses cast upon the cylinders, and applied in combination with means for efficiently preserving the alignment of the several cylinders. In most cases the after cover of each cylinder should be arranged for convenient accessibility to the respective piston. If any special difficulty arises in this the front cover may be substituted. This is, however, always less convenient, chiefly for the reason that the piston cannot be withdrawn by simply uncoupling the piston-rod from the cross-head.

Framing for horizontal engine always subject to bending.—It is practically impossible to arrange the framing of a horizontal engine so as to observe the conditions set forth for observance in vertical inverted engines. A very slight amount of bending action must always remain, even with the strongest bed-plate, and the

utmost degree of security in connection with the foundation. But when the modern type of framing is adopted, and the bed-plate is of good depth, and the piston-rod near to the level of the upper surface of the bed-plate, the bending can never reach objectionable proportions.

Separate framing for two-crank horizontal engine.—In horizontal engines of large size, with two cranks on opposite ends of the crank-shaft, two bed-plates are almost invariably used, quite distinct from each other, and unconnected except through the medium of the foundations and groundwork. It is therefore necessary that the foundations should be connected, with a view to prevent unequal settlement. Templates are necessary for use in setting the two parts in proper relation to each other.

Arrangement of holding-bolts.—Engine-framing of all classes requires to be held to the foundation by holding-bolts, to prevent or reduce vibration and movement. Where practicable, these should be arranged at intervals not exceeding 6 feet apart. A bolt should be placed against each side of each crank-shaft bearing, where the vibration and the liability to move are the greatest. The distribution at other parts is not subject to rule. Lewis bolts are sometimes used, but long holding-bolts, reaching well down into the stone or concrete foundation, are preferable, and are almost universally used. A hand hole is provided to allow access to the lower end of each bolt, and is occasionally convenient in permitting the bolt to drop a short distance.

Adjustment and packing of bed-plates in position.—Perfection of bedding upon the foundation is a most essential point in connection with bed-plates of all kinds. When the upper surface of the foundation is of ashlar stone, the bed-plate is often rubbed to and fro to mark

the stone, then removed, and the stone dressed off by the marks and the help of a straight-edge and level. The stiffest of hollow castings is much more flexible than is generally supposed. This causes the rubbing process to be very misleading at times, and a much better plan, often available where the area of base and the stiffness of bed-plate preclude the possibility of disintegration of cement, is to dress off the foundation with all convenient accuracy about three-eighths to half-an-inch below the level of the bed-plate base. The bed-plate may then be placed in position, accurately adjusted by wedges or forcing-screws, and the space between the bed-plate and the foundation carefully filled with cement, either neat or with a small proportion of sand. The cement should be well fed in by the unremitting use of a wood rammer. The distribution of the supporting wedges requires great consideration, so that each one is placed to receive the weight with a minimum tendency towards bending the bed-plate. It is not necessary or possible to load each support to the same extent. The crank-shaft of an engine may often with advantage be placed in position before completing the adjustment of the bed-plate. The journals should be smeared with raddle, and the shaft turned round and lifted out to ascertain the degree of uniformity of the bearing. This is of great use in combination with other tests, but no test should be omitted to ensure truth in every part. The crank-shaft should be loaded to its full working load, so that it will attain its full working deflection. In this operation no difficulty will be encountered, if the bearings are fitted with brasses to withdraw on slightly lifting the shaft. Other means will suggest themselves in special cases.

CHAPTER XXX.

FLY-WHEELS.

Fly-wheel required to supplement regulating action of governor.—A governor is unable to do more than regulate the number of strokes accomplished per minute by a reciprocating engine, and a fly-wheel is required to approximately equalize the motion in the several parts of the stroke. This regulation is effected by the provision of a heavy mass of material, in rapid motion. This absorbs work when the speed of the engine increases, and restores it when the speed of the engine becomes reduced. Two complete alternations usually occur during each revolution of the crank-shaft.

Regulating action of weight in motion.—A mass of material can be put in motion, or reduced to a state of rest, or its velocity of motion increased or diminished only by the application of force applied through a finite distance. The amount corresponding to any particular operation is considered in foot-pounds, and is estimated by comparison with the conditions pertaining to falling bodies of which full data are known. One foot-pound is the amount of work which is required to lift one pound through a height of one foot. The total number of foot-pounds in any case is found by

multiplying the force in pounds by the number of feet through which it is applied, whether the quantities be whole or fractional. This is true whether the work be done in a vertical direction, in opposition to the force of gravity, or in any other direction, in opposition to resistance of any kind. When work is done in opposition to gravity, and a body of finite weight is raised, the work expended or imparted may be restored in the actual performance of mechanical work during the descent of the weight; or the weight may be allowed to fall freely, which it will do with an accelerating velocity. In the latter case the ultimate velocity will vary with the height of fall. By reason of the velocity thus acquired, the fallen body will perform a total amount of work in overcoming such resistances as may be presented to it, which work will be precisely equal to that expended in raising the weight in the first instance. Of such work, an appreciable amount is due to the resistance of the atmosphere. If the motion is suitably directed, the energy due to the acquired velocity will suffice to carry the body to the height from which it fell in the first instance, less by a small amount due to the resistances.

In a revolving wheel, the position of the centre of gravity of the whole, being in the axis, remains in the same position, so that the action of gravity is neutralized, or no work is done in opposition to gravity. Assuming the whole of the weight of the wheel to be concentrated in the rim, and the wheel to be set in motion by the application of external force equal to its weight, it would assume the same velocity in a certain time as would the same matter formed into a compact mass and allowed to fall freely in space. The energy or work stored within it, stated in foot-pounds, would be equal to its weight in pounds multiplied by the distance in feet

through which it has travelled under such influence equal to the virtual height of fall. If the force applied is greater or less than the weight in any given ratio, the work stored by reason of the application of such force over a given distance will be greater or less than before in the same ratio, and the velocity generated will be greater or less than before in proportion to the square root of the ratio which the applied force bears to the weight of the body.

Variation in production of work by steam during one half revolution.—When steam is applied to a piston at full pressure throughout the stroke, the work done in the different times varies considerably. Dividing the stroke or half revolution into 10 parts traversed in equal times, it is found that the distances through which the piston-rod is moved—and consequently the proportions of the work done—in the successive periods are: 2·4, 7·1, 11·0, 14·0, 15·5, 15·5, 14·0, 11·0, 7·1, and 2·4 per cent. Also dividing the total length into periods corresponding to the usual divisions of the indicator diagram, it is found that the lengths of time occupied over the several divisions are: 20·4, 9·1, 7·5, 6·6, 6·4, 6·4, 6·6, 7·5, 9·1, and 20·4 per cent. of the whole. In each case the angularity of the connecting-rod would somewhat modify the figures given if absolute accuracy were required. Though work is imparted to the crank-shaft so unequally, it is given off with approximate uniformity. This occurs by reason of variations in speed which take place within narrow limits, so that the amount of work which is given out at one portion of the time exceeds that received in the same time, while at another moment the opposite condition prevails. The variation in velocity just described prevails when steam is supplied uniformly. But when expansive working is resorted

to, and steam is supplied intermittently, the irregularity is increased, and a heavier fly-wheel is required to control the speed of turning within the same limits.

Relative variation under different conditions.—The percentage of variation in speed during the several parts of the stroke exists in different cases in direct proportion to the horse-power affected, also inversely as the weight of the wheel and the number of revolutions, and inversely as the square of the speed of the perimeter. These points are perhaps more clearly shown in Table XIX., in which the first line refers to the fly-wheel of an engine indicating 1250 horse-power, the fly-wheel of which weighs 60 tons, runs at 60 revolutions per minute, and with a velocity of periphery of 5000 feet per minute. The power is assumed to be produced only during one half of the stroke, or two quarters of the revolution, but applied continuously throughout the revolution. This condition is not more unfavourable than that which often prevails in a single-crank engine, but in a double- or treble-crank engine the power is produced and applied so as to give much greater uniformity of turning. The conditions are also much modified by the effect of the inertia of reciprocating parts, on which question reference may be made to Porter's work on the "Richards Indicator." In the table, in lines 2 to 5, the successive quantities as specified are separately reduced to one-half, for facility of comparison. In the last column the percentages of variation from the lowest to the highest speed are given, which will be found to accord with the principles given. Cases 2 and 3 point to the advantage of a heavy fly-wheel in comparison with the amount of work done. Case 4 shows a disadvantage connected with long strokes. Case 5 shows the disadvantage of such a speed—2500 feet per minute—as is often adopted in

TABLE XIX.—COMPARISON OF FLY-WHEELS.

Cases.	Specified conditions.				Dynamic conditions.						Variation in speed.
	Weight.	Revolutions per minute.	Minimum velocity.	Indicated horse-power.	Average force at rim of wheel.	Minimum velocity.	Corresponding height of fall.	Minimum work stored.	Work alternately imparted and restored.	Maximum velocity.	
	2.	3.	4.	5.	6.	7.	8.	9.	10.	11.	12.
	tons.		feet per minute.		pounds.	feet per second.	feet.	foot-pounds.	foot-pounds.	feet per second.	per cent.
1	60	60	5,000	1,250	8,250	$83 \cdot \dot{3}$	107·8	14,488,320	171,875	83·8	0·6
2	60	60	5,000	625	4,125	$83 \cdot \dot{3}$	107·8	14,488,320	85,938	83·6	0·3
3	30	60	5,000	1,250	8,250	$83 \cdot \dot{3}$	107·8	7,244,160	171,875	84·3	1·2
4	60	30	5,000	1,250	8,250	$83 \cdot \dot{3}$	107·8	14,488,320	343,750	84·3	1·2
5	60	60	2,500	1,250	16,500	$41 \cdot \dot{6}$	26·95	3,622,080	171,875	42·64	2.3

connection with spur gearing, as compared with the speed of 5000 feet per minute, which is not very excessive for belts or ropes.

Variation in speed not confined to one half revolution.—Another office performed by a fly-wheel is in meeting fluctuations which suddenly arise, either in the supply of steam or in variations in load. These are in all cases ultimately met by the controlling action of the governor, but an appreciable length of time must elapse before this takes effect. In Case 1, Table XIX., if the whole of the load were removed suddenly and absolutely, while the power were continuously applied, the speed of the engine would increase 2·3 per cent. in one revolution. Different conditions of weight of wheel, speed, or horse-power controlled by the wheel, would affect this result, as in Table XIX.

Conditions in construction.—In the construction of a fly-wheel, the chief points to be regarded are—secure connection to the shaft, strength to transmit the power to the rim of the wheel, and safety in work as regards bursting by centrifugal force.

Fixing to shaft.—Large fly-wheels, and heavy toothed wheels and pulleys for belts or ropes, may be all treated together, as to the methods by which they are secured to their shafts. In almost every case the most perfect connection is obtained by staking, for which purpose a central hole is cast in the boss of the wheel or pulley, of larger diameter than that of the shaft at the part which carries the wheel. At this part the shaft is made larger than elsewhere, or is "bossed." This facilitates the planing of the key-flats, and provides greater stiffness of shaft and greater leverage to the keys, by which their tendency to slip in work is equally reduced. A very good minimum diameter of boss $= 1\frac{1}{8}$ diameter of shaft $+ \frac{1}{2}$ inch. The width of key

may then $= \dfrac{\text{diameter of boss}}{4} + \tfrac{1}{2}$ inch, and the thickness $=$ width $\times$ ·4. The shaft and the hole are primarily circular in cross-section. The flats on the shaft are not sunk below the surface, and are parallel throughout. The hole in the boss is for the most part left rough, but is slotted to suit four keys, which are tapered in thickness, but parallel in width. The points of two adjacent keys are directed towards one side, and those of the remaining keys towards the opposite side. Before the keys are made, the wheel is accurately centred on staking wedges, being repeatedly turned into different positions, and adjusted until found correct. Wooden patterns are then made for each key, the keys planed to patterns and fitted by filing. At the edges and ends each key should bear with not less pressure than near the centre. A uniform bearing would be best if it could be obtained with certainty; but when the hardest bearing is near the centre, a rocking action is liable to become developed, after which the work will never be secure until re-fitted. On the whole, it is better that the keys should be slightly recessed along the centre both in length and width.

Strength of boss.—The boss must be sufficiently strong to withstand the keying necessary to secure it to the shaft. The holding power of the keys upon the shaft depends upon the frictional contact, which again depends upon the pressure imposed upon the surfaces. Assuming that the whole force is equally divided over four keys, and that the co-efficient of friction is 0·10, the necessary pressure to be exerted in a radial direction—as at *a* in Fig. 79—by each key, in tons $=$

$$\frac{\text{Indicated horse-power} \times 140}{\text{Diameter over flats of shaft in inches} \times \text{revolutions per minute}}.$$

This tends to burst or split the boss at *c* and *d*, Fig.

79, and is resisted by the combined section at the two points. The key at *b* is the counterpart of that at *a*. Those at *c* and *d* tend to burst the boss at *a* and *b* in a precisely similar manner. A bending action is developed, tending to straighten each of the four quarters of the boss, and the keys are liable to be unequally or excessively driven. The sectional area of the boss should therefore be so proportioned as to avoid the imposition of a greater tensile stress upon the cast-iron than 1 ton per square inch of sectional area, allowance being made for such wrought-iron strengthening hoops as may be provided in the particular case. The area provided should not include any metal which is situated

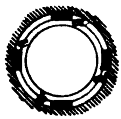

Fig. 79.—Set of staking keys.

at a greater distance than 5 inches from the key-beds or the bore of the boss; as a rule this cannot give any material assistance before the inner portion is strained to the point of danger.

Wheels bored to fit shaft.—Small fly-wheels which are cast solid or in halves, and also small toothed wheels and pulleys, are often bored to fit the shaft and secured by one sunk key, or by two keys set 90° apart. When this is done, it is necessary to take adequate precautions against inaccuracy of any kind, as it is practically impossible to correct such at a subsequent stage. In such cases the best results are obtained by bossing the shaft and cutting the key-bed upon the boss, either as for staking, or otherwise as a sunk groove. The wheel boss is slotted, and the keys should be well fitted all

over. Considered in the abstract, such keys should fit tightly on each side for driving power, and less tightly on the top and bottom, to avoid excessive bursting strain on the boss of the wheel. Practically, however, it is found best to carefully fit the key all over, and to give it a taper in the same direction as one of the keys of a staked wheel, so that the chief pressure is in this case also exerted in a radial direction. This preference is connected with the difficulties which arise in providing driving taper—or "draft"—in the width of the key; from the small area of side bearing in the groove in the shaft; and from the rolling action which is apt to become developed. A parallel key is, however, most efficient for light work, where it is sunk in the shaft, and secured by countersunk screws, then the wheel slipped over it and fixed by a loose collar, or if at the end of an overhanging shaft, by means of a washer and set-screw.

Bosses made in halves.—In some cases a fly-wheel is required to be made in halves, in order to avoid the necessity for the removal of the shaft, or for convenience in transit. In very rare cases the boss of a built-up wheel is so required. Splitting should, however, only be done where unavoidable, when bolts and hoops must be provided of strength sufficient to resist the maximum bursting action of the keys. This can be approximately calculated, as in the last case, or estimated by analogy, from cases in which solid bosses have been burst by driving. As a rule it will, however, be found best to provide all the strength which can be conveniently done. All bolts should be turned and faced, also driven into holes which have been accurately bored and faced. Hoops should be soundly forged, then annealed, and afterwards bored. The beds or shoulders to receive the hoops should be turned to such a dia-

meter as to give a safe strain upon the hoop when shrunk in position. If the boss might be assumed to be absolutely incompressible, a wrought-iron hoop, whose inside diameter, as bored, is $\frac{1}{900}$th part less in diameter than the turned surface upon which it fits, would, after heating to place in position and subsequently cooling, be exposed to a stress of 4 tons per square inch in sectional area. If the diameter were 18 inches, the difference $= \frac{18}{900} = 0{\cdot}02$ inch. The hoop would then require to be heated 180° F.—or a little more for the sake of safety—before placing in position. On account of the slight compression of the boss which takes place, it is found that the allowance for shrinkage may be safely taken at from $\frac{1}{700}$th to $\frac{1}{800}$th of the diameter, and a corresponding increase made in the temperature to which the hoop is raised before placing it upon the boss. If the hoop is made of material of such quality as to safely bear a higher tensile strain than 4 tons per square inch, a larger allowance may be made for shrinkage. If a hoop is made smaller than indicated by these proportions, its safety in use will depend upon its ductility, it will become more or less stretched permanently, and its utility will not be increased. The joint between the two halves of a split boss or wheel may be either planed to a good fit, or it may be obtained by the use of splitting-plates or cores in the mould, in which case the whole boss is cast together and afterwards split. In replacing the halves the utmost care must be taken to ensure perfect contact between the split surfaces. Though this may be efficiently performed, the wisest course usually consists in the adoption of the planed and fitted joint. Hoops to reinforce a solid boss are also often necessary to impart strength to resist the strain due to driving the keys.

Fitting of arms in boss.—The boss is prepared to

receive the arms in bored sockets, as shown in Fig. 80. The total depth of socket is of importance, as affecting the driving leverage and the rigidity of the whole. The actual boring of the socket may, however, be interrupted in the centre part, so long as a good bearing is provided in each end part. Formerly the whole of the housing part of each arm was turned to a taper, and the

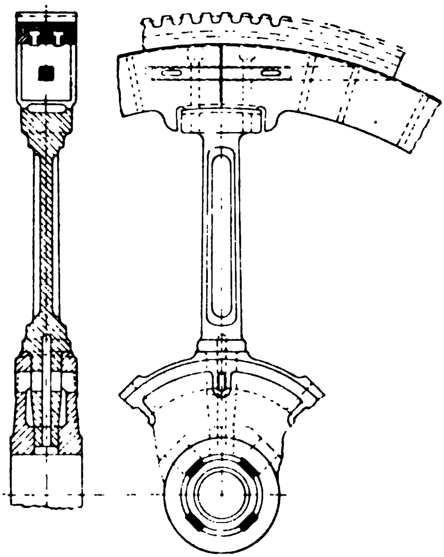

Fig. 80.—Fly-wheel, with spur-segments mounted on fly-rim.

bore made to fit. This gives a good strong arm without weakening the boss too much near to the centre. It is, however, a costly arrangement to construct, and the same object is secured by making the two bearing parts of the arm of different diameter, but each parallel in itself. This parallelism is, however, not absolute, and though the arms require driving all the way, they become more tight as they approach their final positions.

Collars are provided to drive against, but if the arms are fitted with sufficient tightness, the collars do not take a heavy bearing. Single or double-folding cotters are used to fix the arms in the boss. These should have rounded edges, as being less liable to cause breakage of the arms by over-driving, an accident which sometimes occurs and remains undiscovered. But whether the edges of the cotters are round or square, they should be well fitted, so that ample bearing surface is secured, otherwise they are very apt to work loose.

Construction of arms.—The arms of a fly-wheel are almost invariably made of cast-iron, to fit the boss, and of length suitable to reach the rim. There is a great choice in the form of cross-section, but on the whole an approximate H provides the best strength, facility in manufacture, and appearance. The outer end of each arm is finished with a flange or palm, of size sufficient to receive four bolts for attachment to the rim, or in small wheels only two bolts. The outer ends of the arms may be surfaced in the lathe, on the same centres as used for turning the spigots to fit the boss. This course, however, involves a large amount of hand labour in fitting the segments to the arms, even if the whole operation is not confined to chipping and filing. In most cases, however, the whole of the arms are fixed in the boss, and the outer ends afterwards turned off in a lathe or turning mill, especially if means are available for machine facing the inner surfaces of the segments.

Weight segments.—In a fly-wheel for spur driving the fly segments used for weight are attached to the arms by bolts. Usually, these bolts are arranged with heads fitting in T-slots, but in some cases the holes are carried through the segments and the bolts inserted from the outer surface. As a rule, the bearing surfaces

for these bolts cannot conveniently be machine faced, but they should be carefully fitted by hand where necessary. The segments are of dimensions suitable for the provision of the requisite amount of weight, and of length sufficient to reach from the centre of one arm to the centre of the next. The bearing surfaces upon the arms are usually fitted by hand, but in some cases are fitted by machinery, in either case the fitting being confined to narrow strips. The abutting surfaces are often fitted by hand upon narrow strips. Each pair of adjacent segments is secured together by means of dowels and cotters, which also should be fitted the same as those holding the arms in the boss. The centrifugal force imposes an outward tension upon the bolts, and also upon the dowels. The total amount of the centrifugal force can be accurately estimated upon the weight of the segments and of all parts attached to them. It is, however, practically impossible to determine the proportion of the whole force severally sustained by the bolts and by the dowels. The matter is one of vital importance, and therefore the only safe policy is to make each sufficiently strong to bear the whole strain, in case the other should entirely fail to take any. If the segments are slightly over length, the bolts will have some original tension imposed upon them, and though this should never arise, it is wise to give some margin of strength to meet such contingency. The radius upon which the centrifugal force is calculated may be taken with practical accuracy to be equal to the distance from the centre of depth of the segment, including tooth segments, if these are fixed to the fly segments. The tension in tons on each bolt due to centrifugal force =

$$\frac{\cdot 00034 \times \text{total weight (tons) of rim of wheel} \times \text{radius in ft.} \times (\text{no. of revols. per min.})^2}{\text{Total number of bolts in segments}}.$$

The bolts are often fitted with lock-nuts, but these are practically useless when applied in the ordinary way and exposed to any vibration. The two nuts can only prove useful when they are actually "locked," so that they bear hard against each other, and one bears against the forward side of the thread, and the other against the after side. Under such conditions, the outer nut and the part of the bolt connecting the two nuts must bear the strain due to locking, in addition to that of the useful load upon the bolt, while the thread of the inner nut bears only the locking strain acting in the opposite direction, the nut itself acting as a distance-piece under compression. When two nuts are really locked together and the bolt breaks, it usually happens at the point of contact between the nuts, if the bolt is sound: and when the thread of such a bolt wears or strips, it always happens in the outer nut. Consequently when one nut is thicker than the other, the thicker nut should be placed outside, but as the point is subject to great misunderstanding, the only safe plan is to make each nut of such thickness as to bear the whole strain. In all ordinary cases, this is secured by making the thickness of each nut equal to the outside diameter of the thread, and greater in exceptional cases. The strength of the bolts must also be sufficiently great to allow for the application of such a pressure as will retain the surfaces in forcible contact, and by the action of friction will prevent movement. In some cases the actual driving power of the wheel depends only upon the frictional contact of the arms and segments, and the total number of bolts must also be prepared to jointly apply a force in tons =

$$\frac{220 \times \text{indicated horse-power}}{\text{Velocity in feet per minute at ends of arms}}.$$

About half the amount of last item should also be allowed in addition, on account of unavoidable irregularity in tightening the bolts. In the majority of cases, however, the segments are provided with well-fitted snugs, with or without the interposition of keys fitted against the arms. These take the strain arising from the transmission of work, but the surfaces still require to be held, as a precaution against vibration, for which about half as much strength should be provided as is necessary, on account of the driving power of the wheel in the absence of snugs. When the connected segments are faced in a lathe, for bedding on the ends of the arms, the snugs cannot be carried across the full width of segments; but short snugs, well fitted, are quite sufficient. In some cases, for the sake of appearance, the fly segments are recessed, so that the ends of the arms finish fair with the inside of the fly segments. With the same object, the nuts of the bolts which secure the toothed segments are recessed below the surface. Both measures should, however, be avoided. In the first, the bolts and the dowels are crowded together too closely; and in the second case, the proper tightening of the bolts is attended with difficulty, while the locking of double nuts becomes impossible. Dowels were formerly made to follow the curve of the segments, but are much better if made perfectly straight. Two adjacent joints require to be prepared, so that the dowels may be housed in the whole length, to allow the last segment to be placed in position, after which the dowels are brought into position for cottering. The position of such recesses should be clearly marked outside the segments, so that any person may find them at a future time. But a still better course is to cast a long hole in each segment, so that any dowel may be easily reached at any time. Weight and toothed segments

are sometimes most efficiently connected by means of recessed hoops shrunk over each joint.

Toothed segments.—Toothed segments are made of such a form as to give the nearest possible approach to uniformity of thickness in all parts of the casting. If a large mass is provided below the teeth, sound castings are practically unattainable. The teeth of many fly-wheels are, by reason of defective form, exposed to shock, vibration, and irregularity and excess of pressure utterly disproportionate to the work performed, which subject is treated in another chapter. But even where every care is taken to avoid these defects, it would be unwise to relax any effort to secure good sound castings. In the majority of cases, the toothed segments are fixed to the outside of the fly segments. A good arrangement is one in which two T-grooves are cast along the whole length of each segment, and bolts are distributed along the segments as required. The consecutive segments are disposed so as to butt midway between the lines of the arms, but are not connected together in any way, except by means of the fly segments. The driving is then effected entirely by frictional contact, and the calculation for the bolts is the same as for fly segments without snugs, with the exception that only a portion of the bolts in the entire circumference can be depended upon to transmit the working power of the wheel. Security against centrifugal force must also be ensured. Great security is also afforded by the judicious use of shrunk hoops. Many years ago wheels were often constructed in which one set of segments was made to perform the double duty of providing weight and supporting the teeth. It was, however, found to be impossible to obtain good sound teeth in this way. Recesses were tried for lightening the segments, to be afterwards filled by separate weights, but these failed almost equally with the original

form. The number of arms is always even, and in ordinary cases is so arranged that the length of segments of either kind shall be from 6 feet to 7 ft. 6 in.

Toothed segments of reduced diameter.—In order to prevent the gearing speed from exceeding low limits, and still to provide a fly-wheel of superior efficiency, the tooth segments are often made of smaller diameter than

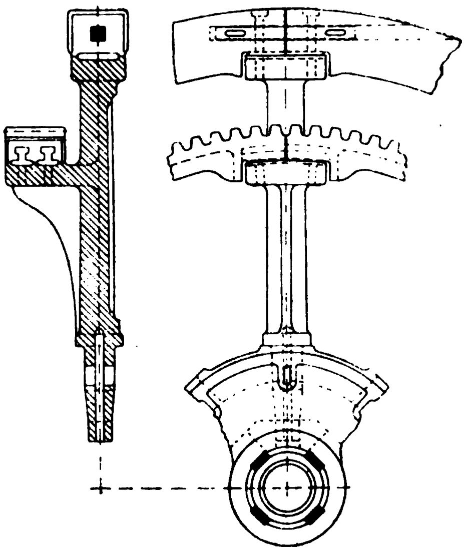

Fig. 81.—Fly-wheel, with spur segments of reduced diameter.

the fly segments, as shown in Fig. 81. The fly segments are made practically as though required to carry toothed segments, with the exception that preparation for the purpose is omitted. They are attached to the arms by bolts, and to each other by dowels or recessed rings, precisely as in the former case. As the outer surface of the fly segment is exposed, the dowel holes, if continued

to the surface, should be blocked by wood or iron stoppers, driven in and securely bolted. The toothed segments are only supported or secured at their ends, and are cast with stiffening ribs along their length. The arms are cast with palms on one side; as these overhang the arms and are exposed to great strain, the arms should be of greater strength than when the toothed segments are upon the central plane of the wheel. This additional strength is of equal importance throughout the length of the arm. The portion next to the boss has to transmit the power from the shaft to the teeth of the wheel, and the portion beyond the teeth has to connect the teeth and the fly rim for transmitting work from the latter to the former. Unless the pinion works overhanging, the diameter of the fly rim cannot exceed the sum of the diameters of the tooth rim and the least pinion, less by the diameter of the pinion-shaft. But subject to this limit the expedient is often of value.

Spur-wheel separate from fly-wheel.—The same object is sometimes secured by providing a spur-wheel on the shaft, separately from the fly-wheel, the connection being made by means of the keying of the two wheels upon the shaft, which then requires to be performed with exceptional care. In such cases a large diameter of shaft is important, as the keys then act to greater advantage, and an exceptional degree of stiffness is secured. If the two wheels are very close together, it will be necessary to plane eight flats on the shaft, so that the head and point of each key is freely accessible by means of a drift. For the same reason, the hole through each boss should be sufficiently large to take the drift.

Belt and rope driving-wheels.—The fly-wheels used for rope and belt driving are constructed on the same

general lines as those used for spur driving, but being usually of larger diameter and much greater width, the whole proportions are changed. In many cases also, the width is so great that a second set of boss and arms is provided. In most cases, however, only one set of segments is employed, the outer faces of which are prepared to receive ropes or belts, as the case may be. Owing to the large dimensions of such wheels, ample weight is usually secured by the adoption of suitable proportions, without providing segments whose sole office is to furnish weight, as must be done when spur driving is adopted. The bosses and arms of such wheels are practically identical in arrangement with those provided for spur fly-wheels. The segments are also secured to the arms in a very similar manner. The several segments are attached to each other by flanges and bolts, the flanges being proportioned with a view to secure the utmost possible uniformity of strength. These joints are generally placed over the arms, in which position they cause a minimum of interference with the strength of the rim, but it is impossible to insert bolts opposite the ends of the arms for connecting the segments together. Sometimes the joints are placed midway between the arms, where they are freely accessible for bolting the whole way across. They are, however, here placed at a disadvantage, by reason of discontinuity of strength and the loading of the centre of the span, by reason of centrifugal force due to the weight of the flanges and bolts. As a rule, the former plan is to be preferred, but much depends upon the proportions of the wheel and the disposal of the metal. In all cases, the strength of the rim against bursting by centrifugal force should be amply provided against. All joints should be connected directly by bolts or other equivalent means, whether situated upon the arms or not, and no

dependence placed upon connection by way of bolting to the arms. The tension upon the total width of the joint =

$$\frac{\text{Weight per lineal foot of rim in pounds} \times (\text{velocity in feet per second})^2}{32 \cdot 2}.$$

A suitable allowance should be made on account of unsupported joints or other irregularities. Messrs. Combe, Barbour, and Combe, of Belfast, make rope wheels in which the segments and arms are together secured to the boss by means of a long bolt passed down each arm. The head of each bolt is recessed in the rim, and the point reaches into the boss, where it is secured by cottering. The entire bolts throw a compressive strain upon the arms, and hold the whole wheel together with security. Such bolts are less liable than cast-iron arms to be broken in driving the cotters.

Wrought arms.—Fly-wheels for rope and belt driving are sometimes made with various combinations of rolled iron or steel plates, flat bars, angles, or T-bars to form the arms. In some cases, these are secured by bolting, and in others by riveting. Perfectly efficient results can be obtained in this way, but transverse stiffness must receive special attention. Wheels of comparatively small diameter may be plated over each side, as described in connection with "pulleys."

Provision for auxiliary turning of fly-wheel.—When all large wheels were made for spur driving, the teeth were found to be convenient for the application of a long iron bar for forcing the wheel round by hand for any purpose. In other cases, holes were made in the fly rim, or other means adopted. On the introduction of belt and rope driving, all former expedients became either impossible or attended with great difficulty. In connection with large engines, this difficulty led to the introduction of small auxiliary engines for barring.

These are usually arranged to drive by a train of wheels, the last pinion of which can be placed in gear with, or withdrawn from, a spur-toothed belt on the main fly-wheel. The withdrawal from contact is done automatically, when the main engine moves ahead by its own steam, while the barring engine is in gear. By this means the main engine can be started from any position, and with extreme slowness, so as to give warning throughout the works. The convenience afforded for examination of the ropes, belts, or other work is also very valuable. The spur belt is usually placed inside the wheel to save width, but is sometimes placed outside. As it is used only at a very slow speed, great strength of tooth is unnecessary. But the stress imposed is so great that the proportions of tooth usually adopted for the purpose would prove quite inadequate, if they were not well rounded in the roots and at the points.

Enclosure of arms of fly-wheel.—At a very moderate speed, fly-wheels cause a considerable disturbance in the atmosphere by the beating action of their arms, and this would become a serious matter at high speeds, leading to great loss of power due to the energy transferred from the fly-wheel to the air, and also loss of heat from the engine itself, caused by the circulation of air. The difficulty is met by the use of boarded surfaces, which may enclose the arms and present only smooth surfaces to the outside air, free from all projections. The chief part on each side is a plane or slightly dished surface, which in some cases is surrounded by a strip of conical form against the rim, to cover the arm ends. The whole should be planed off smooth, and painted or varnished, as well as for efficiency as for appearance. In other cases, the boarding is inserted between the arms, so that nearly half of the substance of each

arm projects beyond the boarding for the sake of appearance, but at some sacrifice in efficiency. Sometimes sheet-iron is used instead of wood, but it is apt to lose shape, and possesses no advantage. Small lugs cast upon the arms are convenient for securing the frames of wood upon which the boards or sheets are fastened.

CHAPTER XXXI.

CONDENSERS.

Purpose of condenser.—A condenser is required for changing the physical condition of steam, after its use in the engine, from the condition of vapour to that of liquid water, whereby its volume becomes several hundred times smaller than before. The temperature of the water discharged from the condenser may not be much below that of the steam as it leaves the engine, but a large amount of heat pre-existing as latent heat of vaporization is surrendered in the operation by the condensed steam. This heat is acquired by water, which for the purpose must be supplied in sufficient quantity and at a sufficiently low temperature, so that the resulting temperature shall be below that of steam at the pressure existing in the condenser. The fall in temperature should be as great as can be practically secured.

Duty of condenser measured by quantity of heat treated.—In another chapter it is explained that 10 to 20 per cent. of the total heat supplied to an engine usually becomes extinguished by conversion into work, so that the sum of the sensible and latent heat of the steam is less after the performance of work than before.

Also that almost the whole of the remaining 80 to 90 per cent. of the total heat is sent to waste through the condenser, or directly into the atmosphere. All calculations with reference to the proportions or the duty of a condenser were formerly based upon the unstable foundation of nominal horse-power. At the present time they are usually based upon the indicated horse-power of the engine, which is an important advance upon the use of the nominal horse-power. But the only true basis is that of the work performed. For the sake of comparison, two engines may be considered, each of which may be described as "excellent," but which, working under different conditions, consume respectively 15 and 12½ pounds of steam per indicated horse-power per hour. The latent heat of about 2½ pounds is required to furnish the heat which is transformed into work, leaving the latent heat of 12½ and 10 pounds of steam respectively as the amount which the condenser must be able to extract from the steam in order to effect its condensation. Assuming that a condenser is adopted which is able to efficiently condense 12½ pounds per horse-power, but that only 10 pounds are required to be condensed, it follows that the condensing power is 20 per cent. in excess of the necessities of the case. Such a condenser will be unnecessarily costly and bulky, but not otherwise objectionable; while in all probability a slightly better vacuum will be secured. A condenser whose proportions are based upon indicated horse-power, applied to practice in which a low steam consumption prevails, will err upon the safe side. When in exceptional cases a condenser is required for an engine whose steam consumption is high, there is, however, some probability that deficient proportions may be adopted. The general tendency of practice being, however, towards a reduction

of steam consumption, difficulties of this kind are very rare. The exact amount of heat to be transferred from the steam to the water of the condenser depends primarily upon the weight of uncondensed steam supplied to it from the low-pressure cylinder, and upon the latent heat of steam corresponding to the pressure, which, for purposes of mental calculation, may be taken to be 1000 units per pound. A further amount should however, be usually added, on account of the fall in temperature from that of the steam at exhaust pressure to that of the condensed steam discharged as water from the air-pump.

Jet condenser.—The jet condenser usually adopted for stationary engines consists of a large cast-iron vessel into which a constant stream of cold water is allowed to flow, under control of an injection-valve, and dispersed more or less completely throughout the space in the condenser. In some cases a perforated rose is fitted to the end of the pipe, or slits provided in it through which jets or films enter at a high velocity, due to the external pressure of the atmosphere. These arrangements are quite efficient, if the whole internal volume of the condenser is filled with the spray, and if the jets are directed downwards, so that they coincide with and do not baffle the current of exhaust steam. In some cases such jets or films are directed upwards, so that the incoming steam must perform work in deflecting or overcoming the force of the jets. In such cases the pressure in the condenser must be greater above the jets than below, and therefore the engine fails to receive full benefit of the vacuum which actually exists in the lower part of the condenser. More frequently a plain pipe is adopted, with an open end turned upwards, and arranged to give a level edge all round, and no further measures adopted for spreading the water. This should

be made with a slightly overhanging lip, to prevent the water from following the outside of the pipe. If the condenser is of sufficiently large diameter to allow the mouth of the pipe to be spread to a trumpet shape, it is all the better. In this way the water does not actually reach every corner of the condenser, but in falling it carries the steam along with it, so that no stagnant air or vapour need be allowed to accumulate. The coincidence of the streams of water and steam is carried to the fullest extent in ejector condensers, of which various patterns are adopted by different makers. Modified ejector condensers are also adopted for use in connection with double-acting air-pumps, or two single ones. In a tall condenser the water may be allowed a long fall, so that it acquires a moderate velocity and becomes finely divided, in both of which ways intimate contact with the steam and efficient condensation of the latter are promoted. When a condenser of the ordinary pattern is adopted, the diameter of chamber should be so proportioned as to bring the water and steam into contact, either directly or in consequence of the intermingling which arises during motion—in this case downward motion. The exhaust-pipe from the engine should be gradually enlarged to meet the diameter of the condenser without abrupt changes. By this means the velocity of the steam is reduced with the least possible loss of energy, and therefore with the best result upon the vacuum.

Presence and action of air in condenser.—The steam passed into the condenser from the engine carries with it a quantity of air. This is primarily due to the air dissolved by the water on exposure to the atmosphere previous to its entry into the boiler. Under ordinary atmospheric conditions this amounts to about one-fiftieth part of the volume of water. When, however,

it is partially relieved from pressure, as in the condenser, its volume is increased. At an absolute pressure of $2\frac{1}{2}$ pounds per square inch its volume will be six times greater than at atmospheric pressure, or about one-eighth part of that of the liquid water in which it was dissolved. The condensing water also yields air in the condenser in the same proportion, so that the smallest proportion of air which can ordinarily exist in the condenser is equal to about one-eighth part of the volume of water discharged. This quantity may, however, be very much increased by reason of inward leakage of air at the stuffing-box of the low-pressure piston-rod, or at the various joints in the exhaust-pipe, or by a defectively-arranged feed-pump, so that the air-pump should be capable of withdrawing the whole of the water which at any time may be required to be supplied to the condenser, and, in addition, at least one-third of the same volume of free air. One single-acting air-pump only draws the water and air from the condenser during one-half of the whole time, so that during the other half they must accumulate in the condenser. This gives rise to fluctuations in the pressure inside the condenser, which fluctuations cannot be avoided by any arrangement of proportions, but which are minimized by an increase in the capacity of the condenser. This capacity should be not less than from five to ten times the volume swept by the air-pump piston, when this is of minimum proportions. The vacuum will then usually be so maintained that the gauge will oscillate from one to one and a half inches during the time of each revolution. In some cases two single-acting or one double-acting air-pump is used, when the capacity of the condenser may be reduced if desired; but in all cases ample space is advantageous in steadying the vacuum. A very good

about three-eighths to half-an-inch clear from tube to tube, but would be better if spaced a little further apart. It has been found that by opening out lanes, by the removal of tubes, the condensing power is increased. Tubes set in the ordinary manner possess condensing power equal to the absorption of 13,000 units of heat per square foot per hour, when the condensing water is supplied at 60° and allowed to rise to 95° F. before discharge. In $\frac{3}{4}$-inch tubes, 11 feet long, through which the condensing water makes two passages, the equivalent velocity of water through the tubes is about $3\frac{1}{2}$ feet per second. With tubes spaced a little further apart the condensing power may be increased about 20 per cent., but the velocity mentioned ought to be considered as a maximum. The velocity through the tubes which is necessary to secure a given rate of transfer of heat per square foot per hour varies directly as the length of the tube and inversely as the diameter of tube, and amount by which the temperature of the condensing water is allowed to rise. Though 13,000 units of heat per square foot per hour may be easily transmitted through the tubes under favourable conditions, it is wise to make preparation only for the transmission of 10,000 units per square foot per hour. This will give a margin for the occasional use of water at a higher temperature. The latent heat of steam at temperatures prevailing in the condenser is almost exactly 1000, so that each square foot of condensing surface, measured inside the tube-plates and around the circumference of the tubes, is equal to the condensation of 10 pounds of steam per hour, or rather less if the temperature of the condensed water is appreciably reduced after condensation.

Securing of tubes in surface condenser.—The tubes of a surface condenser are held in tube-plates of rolled

muntz metal. Two systems only of holding the tubes need be described, which are of nearly equal merit. Under the first system the holes to receive the tubes are bored through the tube-plates to one uniform diameter, about five-sixteenths of an inch larger than the diameter of the tubes. Ferrules of soft wood are driven to surround the tubes in the holes, and make a tight joint. The ferrules are very dry when inserted, and are previously compressed in a small lever press; afterwards, and by the access of moisture, they swell and become very tight. The second arrangement is more costly. The tube holes in the tube-plates are first drilled clear to the size of the outside of the tubes, after which they are enlarged to about two-thirds of the depth, and prepared to receive screwed ferrules or glands. The tubes are packed by placing a little yarn in each of the stuffing-boxes thus formed, and screwing down the ferrules, which is effected by means of a tool fitting into two opposite notches in the outer end of each ferrule. The ferrules are usually cut off screwed tube, the ends cleaned, and the notches cut. Unless well packed, these are a little more apt to allow the tubes to slip longitudinally than are wooden ferrules. Such slipping arises from the differences in expansion of tubes and condenser shell, due to changes of temperature, and is especially pronounced in vertical condensers, in which the weight of the tube tends to determine the slip in a downward direction. Screwed ferrules are, however, very easily arranged, with the outer ends pressed inwards to form a lip projecting into the bore of the ferrule, effectually preventing slip of the tube. This precaution should be observed in all condensers fitted with brass ferrules, but the lips should not project over the bore of the tube so as to interfere with the current of water passing through. When one

tube or more move longitudinally, so as to drop out of the tube-plate, an opening is left through which some of the condensing water passes into the condensed water and is fed into the boiler. Unless the condensing water is very injurious to the boiler no harm need be apprehended, and the leakage is immediately announced by the rise of the water in the boiler, or the water running to waste from the feed tank. The length of the tube is generally decided largely by the length of condenser which is structurally most convenient for working into the engine framing. Tubes are made up to 15 feet in length, but are better if not above 12 feet, while even shorter lengths may be desirable to reduce the velocity through the tubes while obtaining a high rate of condensation per square foot of surface. When the length of tubes exceeds 7 feet, they should be supported at intervals not exceeding 6 feet by iron plates with smooth drilled or rimered holes to fit each tube. If these are only roughly punched, the tubes will probably become perforated.

Construction of condenser shell.—The condenser casing or shell may be of any convenient shape, to suit the position decided upon. In the great majority of vertical inverted marine engines, the condenser is worked in with the engine framing, the bottom being almost flat,—a slight fall being given for drainage,—the sides are nearly or quite vertical, and the top of semi-circular section. For condensers which are not structurally connected with the main engine, a circular form is usually adopted, placed either vertically or horizontally. All flat surfaces must be made of strength sufficient to safely stand the pressure, with or without supporting ribs. When ribs are adopted they should by preference be placed outside, so as to give the best effect in strength to oppose the atmospheric pressure correspond-

ing to the vacuum inside. But a good plan is to provide ribs both outside and inside. Those placed outside should be arranged vertically to clear connecting-rods, pumps, and other details, while those inside may run horizontally, almost touching the tubes, and preventing the current of steam from straying between the tubes and the shell, so as to escape intimate contact with the tubes. When the tubes are set horizontally, a sufficient amount of space must be allowed above them, so that the steam may spread over the tubes before descending, and the diaphragm plates which carry the tubes must be so arranged as to avoid interference with the distribution of steam throughout the condenser, and freedom of drainage at the bottom. Each tube-plate should be planed inside, and also the corresponding face upon which it is secured in the condenser. When brass ferrules are used, the tube-plates up to a clear width of 24 inches should be $1\frac{1}{8}$ inches thick. But when wood ferrules are used, more material is removed from the holes, and the thickness of tube-plate should be $1\frac{1}{4}$ inches, to provide strength to withstand the pressure. Tube-plates of greater width should be increased in thickness in the proportion $\sqrt{\frac{\text{clear width in inches}}{24}}$, or the plate will spring in use. The tube-plates receive practically no support from the tubes, as these are long and thin, and are too lightly secured to serve such a purpose. The tube-plate must therefore depend on its own strength to withstand the pressure of the atmosphere.

Circulation of cooling water.—The condensing or circulating water is admitted at the bottom of one end of the condenser, passed through the lower half of the tubes, returned through the upper half of the tubes, and discharged from the top at the same end to which it was admitted at the bottom. The cover at this end is therefore provided with a diaphragm to separate the

two sets of tubes, and opposite to this diaphragm the tube-plate is left solid. The covers at each end are arranged to leave sufficient space to easily pass the water. The inlet and outlet branches should in all cases be provided upon the condenser itself, so as to avoid breaking a pipe-joint when removing a cover. Each cover should also be provided with one or two small doors for convenient inspection. These and the inside spaces should be sufficiently commodious to allow the passage of a man's arm. A bye-pass or supplementary feed-cock should be provided, to take a little of the circulating water into the steam space of the condenser near to the top, in any convenient position at either end. This is for the purpose of making up loss of boiler feed-water, however this may arise. The surface condenser of a marine engine is always provided with means for jet injection. This is useful in case of accident to the circulating pump, but is chiefly adopted with a view to keep the bilges clear of water.

General.—A surface condenser casting should be made with the same care as a cylinder. Thicknesses should be made as uniform as possible, angles filleted, and salient corners rounded off. Necessary stiffness may be imparted by ribs fitted as described above. Surface condensers have not been largely applied to stationary engines, but the great advantages which they possess in maintaining the purity of the boiler feed would in many cases amply repay their cost. They are never found to give serious trouble on board ship even in the tropics, where the temperature of the water used for condensation reaches 90°.

Atmospheric surface condenser assisted by water.—A modification of the surface condenser is used in which the steam circulates through the tubes, the heat being abstracted from the steam through the agency of water,

which is allowed to trickle over the pipes. Latent heat is surrendered by the steam inside the pipes to the water on the outside, the latter being raised in temperature and evaporated into the atmosphere during the operation, while the steam within the tubes is simultaneously condensed. Mr. Longridge, in his report for 1892, gives an account of a test made upon such a condenser, fitted with ten inverted U-pipes of cast-iron $4\frac{1}{8}''$ outside diameter, $\frac{5}{16}''$ thick, connected to horizontal cast-iron pipes $10\frac{1}{4}''$ and $7\frac{1}{4}''$ external diameters, $\frac{5}{8}''$ thick, of which the total external surface was 272 sup. feet. Water was supplied by a trough over the pipes. In an experiment made in a drizzling rain without wind, nearly 2000 units of heat per square foot per hour were transmitted. The atmospheric conditions were obviously very unfavourable, and judging from the temperature in the hot-well (136·5°), the appliance may be assumed to be working to its full capacity.

Atmospheric surface condenser unassisted.—A surface condenser may be constructed for exposure to air alone. Such condensers exposed to still air require to be made of enormous area for even very moderate powers. Those made for tramway engines are arranged to take advantage of the motion of the car through the air. Fouche's air condenser is illustrated in *Engineering*, vol. xlviii., p. 246. In this a fan is used to throw a large current of air upon the tubes, which are arranged horizontally. 1,141 square feet of surface suffice to condense 725 pounds of steam per hour, equivalent to a transmission of 640 units of heat per square foot per hour. To effect this purpose 893,000 cubic feet of air are employed, and raised in temperature from 76° F. to 117° F. The vacuum obtained is only equal to 18 inches of mercury. By increasing the volume of air the vacuum is improved. Better results are also obtained when water is distri-

buted upon the pipes, which is equivalent to a combination of the present principle with that last discussed. The transmission is then found to reach 4000 units per square foot per hour. In the above example the fan is driven at 195 revolutions, is 6 ft. 6 in. in diameter, and consumes 2 horse-power. The hot air produced may sometimes be very serviceable, when the use to which it is put will determine the degree of moisture which may be imparted to it. In some cases air exhausted under a system of ventilation may be utilized to good effect by discharging it into an air condenser wet or dry.

Ejector condensers.—Ejector condensers are especially useful when the water is received under a head of 6 to 15 feet, at a low temperature and abundant in quantity. The vacuum is steady and no air-pump is required, though a circulating-pump may be employed if desired to raise the water for the supply of the condenser. The vacuum is, however, rather weaker than with a condenser and air-pump. An ejector condenser consists of a series of cones accurately centred in line. The first or inner cone is supplied with water, and the outer ones with steam from the engine. Intimate contact between the steam and the water is produced by the high velocity at which the former enters the instrument and the form of the cones, the actual combination occurring in the throat or narrow part of the instrument. Sometimes a smaller cone is added inside the directing cone of the water supply; this is for the addition of a little steam for starting or assisting the operation, when the water is supplied under a deficient head. Such an instrument is applied to the inlet of an ordinary condenser with advantage, sufficient space being arranged between it and the air-pump to equalize the pressure; its use usually necessitates the adoption of two single-acting air-pumps or one double-acting.

Advantages attending the use of a condenser.—The use of an efficient condenser of any type increases the range of steam-pressure in the engine, and causes an increase in the amount of power to be obtained from a given quantity of steam. Conversely, a reduction is effected in the amount of steam required for a given purpose. Consequently, less feed-water and less coal are required when a condenser is used, than when the same work is performed without one. A jet condenser is economical in original cost and in cost of maintenance. It is therefore to be preferred when an abundant supply of good water is available. A small amount of water is required to make up for waste. A surface condenser is equally efficient as to the vacuum produced with the last, but is more costly to provide and maintain. It effects a saving in the cost of boiler cleaning, and in the losses due to a dirty condition of boilers. The amount of water required to make up for loss is about the same as in the last case. The use of a condenser of either of these two types is attended by a better vacuum than is secured by other means. A cooling-pond or its equivalent is required equally in either case, unless an abundant supply of cold water can be derived from a running stream or canal. An evaporative condenser is bulky, and much more costly than those just referred to, and is therefore unsuitable for large powers. It is equal to an ordinary surface condenser as to its economical use of good water for boiler feeding, but decidedly inferior to it in consumption of inferior water for condensation, the quantity of the latter required being nearly equal to that of the steam used. An air condenser is still more bulky and costly. It is very economical in water for boiler feeding, and none else is required if the condenser is worked dry. If worked wet, the quantity of heat transmitted will depend largely

upon the quantity of water used, but may exceed the sum of the amounts due to a dry air current and to the use of water in still air. An ejector condenser is the most economical of the whole, in cost and in space occupied. A good water supply is essential to success, and the condensed water cannot be kept separate from the condensing water. An air-pump is not necessary, but the vacuum is only moderately good.

Advantages of low temperature. Vacuum referred to atmospheric pressure.—In every class of condenser a better vacuum is produced in proportion to the reduction of temperature in the steam space, as indicated by the temperature of the water discharged from the condenser. The full average pressure of the atmosphere at sea-level is 14·7 pounds per square inch, which is equal to the support of a column of mercury 30 inches in height, but this varies largely. An absolute vacuum never exists in any practicable condenser, some residue of pressure always remaining. If an absolutely vacuous space be connected to the top of a barometric column, the height of the column will be unaffected. If, however, the latter be connected to the interior of a condenser, the height of the column will become reduced in exact proportion to the deficiency from a perfect vacuum. In other words, the same vacuum will cause an equal reduction in a high column as in a low one. Therefore, any statement as to the measure of the vacuum by comparison with the pressure of the atmosphere is essentially incomplete, unless the barometric pressure of the atmosphere at the same time and the same spot is also given. The pressure at a level of about 900 feet above sea-level is less than that at sea-level, so that the column is about one inch less, but this proportion does not continue quite uniform to greater heights.

Low-level supply of injection water.—The surface of water from which a condenser is supplied may be lower than the condenser by any distance not exceeding 30 feet. It is, however, better that the difference should not exceed 18 or 20 feet. When such difference is great, the pipes for the supply of injection water and the valve for its control should be made specially large, so that the resistance opposed to the passage of the water shall not prevent the pressure of the atmosphere from maintaining a sufficient supply of water.

CHAPTER XXXII.

PUMPS.

Pump passes fluid in one direction.—An ordinary pump is an instrument employed chiefly for the purpose of lifting or forcing water, and consists essentially of a chamber the capacity of which may be alternately increased and reduced, and which is fitted with two valves, each of which allows the free passage of fluid in one direction only. Through one valve the fluid is received into the chamber, and through the other discharged from the chamber.

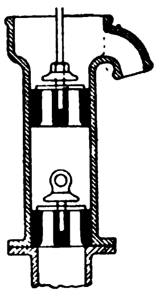

Fig. 82.—Bucket-pump.

Bucket-pump.—The simplest type is that of the plain bucket-pump, shown in Fig. 82, in which a piston

or bucket fits easily in a vertical bored barrel. A suction-valve is provided in the lower part of the barrel. During the up-stroke of the bucket the pressure of the atmosphere is excluded from the space beneath the bucket. The atmospheric pressure acts, however, upon the surface of the water supplied to the pump, and causes it to pass through the suction-valve and follow the rising bucket. During the down-stroke of the bucket, the foot-valve is closed, so that the water beneath the bucket cannot return, but must pass up through the bucket-valve, so that at the end of the down-stroke most of the water is above the bucket.

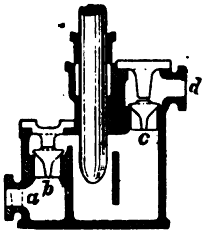

Fig. 83.—Force-pump.

During the next up-stroke the first charge of water is further raised and discharged by the spout, while a second charge is drawn in beneath the bucket, to be passed through the same cycle as the first. A third valve provides additional security, but is not necessary while everything is in good order.

Force-pump.—In the force-pump shown in Fig. 83 a solid ram is used to give the variation in volume of the pump-chamber, an air-tight joint being obtained by means of a stuffing-box. On the withdrawal of the ram, the water is propelled through the suction-pipe *a* and the suction-valve *b* into the pump-chamber; and on the re-insertion of the ram the same amount of

water is driven forward through the discharge-valve *c* and the discharge-pipe *d*. The whole of the pump-chamber below the ram, above the suction-valve and beneath the discharge-valve, is at all times filled with water during effective work, the total amount varying by the volume due to the area of ram and to its length of stroke.

Amount of water discharged.—The amount of water supposed to be discharged by the pump first described is that due to the area of bucket, multiplied by the length of stroke and by the number of revolutions in the time under consideration. That discharged by a force-pump is measured by the sectional area of the ram, multiplied by the length of stroke and by the number of revolutions. At ordinary speeds the quantity of water actually discharged is found to be rather less than calculated as above, the difference being due to "slip" or loss of water at the valves.

Lift or opening of pump-valves.—Pump-valves or clacks are required to open to a certain distance to allow the passage of water, and alternately to close, in order to prevent its return. In the abstract it is very desirable to allow the current of water to flow through the valve-passages and pipes with uniform velocity at all points, to secure which an ordinary valve would require to lift through such a distance as to give a clear opening around the circumference, equal to one-half of the radius, or to give the same area in some other manner. To observe this condition, a valve 3 inches in diameter would require to lift three-quarters of an inch clear. Experience shows, however, that valves thus arranged always knock heavily, unless special means are adopted to prevent it. Assuming that the ram of the pump shown in Fig. 83 is in about the position shown (half-stroke), and rising, the suction-valve is fully open,

and will remain so until the ram reaches the top of its stroke. The valve will then fall to its seat during the first part of the time of descent of the ram, *after* which an appreciable—but shorter—length of time will be occupied in lifting the discharge-valve. At the completion of this time the ram will have advanced well along its stroke, and be moving at a correspondingly high velocity. Assuming that the diameter of the valves is equal to that of the ram, the discharge-valve will at the moment of complete opening be moving at the velocity of the ram, when it will be stopped suddenly by contact with the guard, and a distinct—and often a heavy—blow will occur. The discharge-valve remains wide open until the end of the down-stroke is reached. Its closing occupies a length of time equal to that occupied by the suction-valve in closing, and is followed by a period in which the suction-valve opens, which again is equal to that occupied by the discharge-valve in opening, on the assumption that the motion of the ram is symmetrical, and that the valves are allowed equal amount of lift. If the diameter of the ram is 3 inches and its stroke 12 inches, the diameter of the valves 3 inches, and $\frac{3}{4}$ inch lift allowed, one-eighth of the stroke will be completed before each valve becomes fully open. At such moment the ram will be moving with two-thirds of its maximum velocity, which fully accounts for the blow experienced. If the lift is reduced to one-quarter of an inch upon each valve, the double operation is performed while the ram moves through one-third of the distance, in a little more than half of the time. At the time of complete opening the ram is moving with only four-tenths of its maximum velocity, so that the violence of the blow is materially reduced. Usually, however, three-sixteenths of an inch is quite sufficient to allow such valves to lift.

Loss of water at valves.—Valves which close in the manner just described also lose water to an amount corresponding to the lift allowed. That is, the water beneath the valve when the latter is lifted into its open position is returned and lost, as affecting the amount discharged by the pump. In this way the volume really discharged per revolution = area of ram × (stroke − lift of suction-valve − lift of discharge-valve). Whenever heavy pounding of valves takes place, it may be safely concluded that it is accompanied by loss of water.

Modifying conditions as to loss of water and pounding. The condition just described applies with a close degree of approximation in cases in which pumps of ordinary arrangement are driven at speeds above 50 revolutions per minute. At lower speeds, and especially if a pause is allowed at each end of the stroke, the valves are allowed to fall to their seats at an earlier period. The same result is also produced by spring-loading of the valves, which is achieved in different ways, but preferably by means of spiral springs of steel, placed either inside the chamber, or outside and applied to a small spindle working through a stuffing-box. By either means each valve may be caused to reach its seating about the time the end of the stroke is reached. Each valve then rises much more easily than is otherwise the case, the working of the pump is improved, and the discharge increased. The spring-pressure should be applied with uniformity on each side of the centre. In a less degree, the same result is obtained by the use of heavy valves, which will fall in advance of the water. Pounding of the suction-valve is due, not only to the weight of the valve itself, but of the column of water in the suction-pipe; that of the discharge-valve is due to the column of water in motion in the discharge-pipe.

In either case the violence of the pounding action is mitigated by the provision of a second valve. Thus an ordinary well-pump has a foot-valve inserted in the end of the pipe in the well, and possibly an intermediate valve at a higher level. In like manner the discharge-pipe is provided with a check-valve close to the boiler. If these are ordinary unloaded valves, the amount of water discharged is not affected, but the pounding is relieved by distribution over an increased number of points. Pumps of the Worthington and other types which give a pause at each end of the stroke, allow the valves to fall by sheer weight, to great advantage. Pumps of all kinds are well known to work more easily at low speeds.

Delivery in excess of calculated quantity.—In exceptionally well-arranged pumps, in which the water is passed in a fairly direct line through the pump, without any actual reversal of direction, and which are worked at a very high speed, the quantity of water discharged may somewhat exceed the calculated quantity. Practically, this condition is confined to pumps of the bucket type.

Amount of work done by pumps.—The work done in forcing water to a height above the pump, or drawing it from a lower level, is essentially equal to that which would be consumed in lifting the same weight of material of any kind by other means to the same vertical height. In like manner, the work done in forcing against fluid *pressure* may be estimated in equivalent head or difference in level. The actual pressure which causes the water to rise to the pump is applied by the atmosphere, but the pump is exposed to a resistance due to the full atmospheric pressure upon the pump-ram or bucket, minus the reduced pressure in the pump-chamber, which resistance is overcome by

work which is equal in amount to that which would be necessary for lifting the weight of water in a direct manner plus that due to frictional and eddying resistances.

Possible height of suction.—The height to which water can be drawn, or the maximum possible difference in level between the surface of the water from which the supply is drawn, and the position of the pump-chamber, depends chiefly upon the pressure of the atmosphere. When this is such that the mercurial barometric column measures 28 inches, the water may be drawn to a height of 31·7 feet above the surface level of the supply. When the column measures 31 inches, the difference in level may reach 35·1 feet. In each case a deduction is necessary, on account of the vapour tension of water at the particular temperature. At a temperature of 50° F. the deduction is ·4 foot; at 100°, 2·2 feet; and at 150°, 8·5 feet. But even at low temperatures, it is found that a pump fails to work in a satisfactory and reliable manner when the water is drawn from a level more than 25 feet below the pump, which must be further reduced in the event of high temperatures occurring.

Presence and effect of air in pumps.—Pumps whose office is to raise water are often found to contain air, or air and water. This may arise from the water leaking away during a period of stoppage, chiefly by way of a leaky suction-valve. In many cases, air is drawn into the pump during ordinary work, when it must be passed forward automatically, or the action of the pump will cease. For this reason, there should be no inverted pockets in which air can collect, but every particle should pass upward and onward through the discharge-valve. Air which is imprisoned in the pump-chamber, or in any other position having free communication with the same, is alternately compressed

and dilated by the action of the pump-ram. The result of this is that the opening of the suction and discharge-valves is delayed, and the ultimate arrest of the valve against its stop takes place with abnormal violence. A well-designed pump works quietly and uniformly, but large numbers exist in which the air collects and is passed forward in gulps. Such a pump is at all times noisy, but at times so much so that a general rending only appears to be escaped by a miracle. This defect is often found in the pumps of small high-speed surface-condensing engines, air being introduced by reason of a radical defect in the system elsewhere referred to; but apart from this, the pump should not be subject to accumulation of air, as air is at any time liable to enter in small quantities from leakage at the stuffing-box, or through slight leakages in pipes. Water from almost every source also contains air in solution, which is sometimes liberated in the pump. The pump shown in Fig. 83 is quite free from liability to air accumulation, but in some cases the discharge-valve is set at a lower level, so that air accumulates in the pump-chambers around the ram, the pump becomes liable to stoppage, and is indeed utterly unfit for use under difficult conditions. It can be operated by means of judiciously arranged pet-cocks and close attention, but is always a source of danger and annoyance, though the type is far from uncommon. Water at high temperatures evolves vapour, which affects the action of a pump in a manner very similar to that of air. This defect is occasionally though not frequently met with in connection with bucket-pumps. In some cases, the barrel of an air-pump is continued below the top of the foot-box, in such a manner as to leave a space in free communication with the pump-chamber, and in which air may collect beyond the possibility of discharge, except by means

of special appliances. This air becomes alternately compressed and dilated by the working of the pump causing more or less shock in working, an appreciable loss of vacuum, and general inefficiency. In rare cases a badly-fitted pump-liner leaves a space in connection with the chamber which becomes filled with air, with a similar result.

Velocity through pump-valves.—The valves of a force-pump should be of diameter sufficient to give ample area for the passage of the required amount of water, without requiring an excessive amount of lift in the valve. The shock and loss of water which are thus caused have been already referred to. Feed-pump valves should rarely lift more than three-sixteenths of an inch, and small ones should not exceed one-eighth. The diameters should be such that the average velocity during the half revolution does not exceed 10 feet per second. In many cases, the velocity must be allowed to reach 20 feet per second, but this should only be allowed from absolute necessity. In such cases, the head of the valve should be made flat over a large surface, and an equal area provided in the stop. The water between the two must then be forced out before the surfaces can come into contact, by which resistance is interposed and shock mitigated.

Use of air-vessels.—In connection with an ordinary feed-pump, the water in the whole length of the suction-pipe and in the discharge-pipe is alternately set in motion and brought to rest, with more or less shock. If, however, an air-vessel is provided in connection with the suction-pipe, and a second one in connection with the discharge-pipe, the passage of water through each pipe is equalized. The air-vessels must be placed in free communication with the pipes, near to the pump, and contain sufficient air, so that by its alternate

contraction and expansion water may be alternately received in the vessel and returned into the pipe, and the stream along the pipe rendered more uniform, to the great benefit of pump, pipes, and valves. So long as it remains charged with air, the vessel on the delivery side is most beneficial; but if the water contains little air, it is liable to absorb that in the air-vessel, which then becomes "water-logged" and useless. Owing to the less pressure to which it is exposed, an air-vessel on the suction side is much less liable to this trouble. Cocks should be provided upon air-vessels, for use in drawing off water or admitting air. These should, however, be placed at a low level, so that if they should prove leaky they will not allow the escape of air. Appliances are available for the automatic supply of air to an air-vessel. These should, however, be cautiously used in connection with boiler feed-pumps, owing to the injurious action of air in excess, in promoting corrosion. They are, however, necessary in plant for pumping mines, in connection with which purpose they are described in a paper by Mr. Dugald Baird, read before the Mining Institute of Scotland in 1889.

Pointing of pump-rams.—Pump-rams are usually cut off square in the pump-chamber. Assuming that such a ram could be withdrawn—say an inch—without the water following it, there would be a space of uniform thickness over the area of the ram left to be filled. When the operation is repeated with a ram possessing a conoidal point, the volume of displacement is unaffected, but the space left vacant is distributed over a larger surface. In both cases, in practice, the water closely follows the ram, so that no space is left, but it is evident that the larger amount of surface over which, in the latter case, the same volume is distributed,

causes a reduction in the local disturbance of the water. This distinction assumes great importance in large, quick-running pumps, and is vital to the success of air-pumps, such as those of the Allen engine; in small pumps it is of little consequence, except when worked at a very high speed.

Use of foot-valve in water-pipes.—A pump which draws water from a lower level is liable to violent pounding, to prevent or mitigate which a foot-valve is often provided. Such foot-valve is not absolutely required for any other purpose, though it is of use in case the pump suction-valve fails; and the existence of a foot-valve allows the filling of the pipes and pump above all stuffing-boxes for the detection of leakage, as referred to in connection with centrifugal pumps.

Gradient of pipes.—All pipes for the conveyance of water, especially at a low pressure, should be laid so as to clear themselves of air without attention. As a rule, they will do this if laid with an accurately uniform fall. An excellent plan, possible in some cases, is to begin at the end of the pipe which is nearest to the source of supply, with a rapid descent, to reach a level lower than the discharge end, which is reached by a long rising gradient, in which air is easily carried forward. In other cases, a number of summits may be arranged, each with an open pipe reaching above the level of supply, to remove the air. In all cases, accuracy of level is essential, especially with small pipes. If this fails to receive attention, a number of valleys and summits are formed, in the latter of which air accumulates, which cannot escape, and which interferes with the flow of water. In this manner, many feet of fall are sometimes wasted in a mile.

Water from surface condenser fed into boilers.—

Feed-pumps are usually driven by the main engine, in which case they work at a constant power, which, to provide for emergencies, waste, and blowing-off, must be greater than the average required. The supply is usually taken from the hot-well. In an ordinary jet condensing engine, there is always sufficient water in the hot-well to keep the feed-pump suction-pipe completely supplied. The water is admitted to the boiler under the control of the boiler check-valve, and all excess above that required by the boiler is returned to the hot-well through an overflow valve, loaded—preferably by means of a spring—to a little above boiler pressure. But in an engine fitted with a surface condenser, the amount of water discharged into the hot-well from the air-pump is strictly limited. When the condenser is tight, and the supplementary cock closed, there is not sufficient water in the hot-well to keep the feed-pump fully supplied, and it draws air. Usually the whole of this air is pumped into the boiler, where it may cause much corrosion of the structure. This irregularity of working is also very injurious to the valves and connections, causing great wear and tear. The most effective remedy consists in the exclusive use of the boiler donkey-pump for feeding purposes. This can be adjusted to the work, so that practically no air can be taken into the boiler. A small tank should be provided to receive the water from the hot-well, and it would be better to fit this with a float to regulate the supply of steam to the donkey. The exhaust steam from the latter should be led into the main condenser, chiefly with a view to saving of water. When a surface condenser is adopted, the feed-valve is usually unprovided with an escape-valve to discharge into the hot-well. But the addition of this, or an adjustable bye-pass and valve, will provide the means whereby the suction of

air into the feed-pump may be entirely prevented, to the very great benefit of the boiler. Air may also be separated from water by conducting the whole from the pump into an air-vessel, discharging upward at a high level, and taking out at the bottom. The undissolved air will then collect in the upper parts of the vessel, effecting useful regulation of the stream, and the excess may be discharged under the control of an automatic valve.

Driving of feed-pump.—The ram of a feed-pump may be placed vertically or in any other direction with equal efficiency. This is usually driven at the same number of revolutions as the engine which drives it, but with a shorter stroke. Sometimes, however, a pump-ram is attached to the cross-head of the engine, when its velocity is not reduced, but its area is reduced instead. As regards the pump itself, this is of little consequence, provided that the end of the ram is pointed. But the irregular stress thus imposed upon the cross-head is very apt to rapidly throw it out of truth, unless special precautions are adopted. It should be provided with a stuffing-box and gland, arranged with cups, so that a slight oozing of water may be collected and led away by a pipe. Such cups are also convenient for testing the tightness of the ram in a vertical pump.

Valves and seatings.—The valves and seatings of a feed-pump are of gun-metal, phosphor bronze, or other good mixture. The seatings may be secured either by screwing, or by a tight-forcing parallel fit. They should be sufficiently stout to withstand considerable pounding. The seatings should stand a little way above the metal into which they are driven, otherwise small pieces of grit which may be in the pump will sometimes wash back on to the seating just as the valve drops, and

cause indentation in both valve and seating. Probably, in some cases, cast-iron of hard, tough nature will be found eligible if not superior to gun-metal in use. The majority of valves are arranged with a mitre seating. They are usually made with three wing-guides, which should interfere as little as possible with the free passage of the water. Mushroom or spindle-valves are very seldom used, as they give greater obstruction to the current, and are very apt to break off the spindle. Spherical valves are used most successfully in the pumps of locomotive engines. Each valve should be conveniently accessible for examination on the removal of a valve-cover. Each cover should be held by the least possible number of bolts, for easy removal, and for a suction-valve should be fitted with a projecting part, fitting the hole in the pump casting, with a view to prevent the accumulation of air, as shown in Fig. 83.

Uniformity of wear of valves.—Each valve should be as far as possible protected from any cross current which could tend to throw either the top or the bottom towards one side, and thus cause increased wear in the bore either upon the valve or the seating. This defect is aggravated by excessive lift of valve, and by placing the suction and discharge-branches and the openings into the pump-chamber near to the level of the valve.

Air-pumps.—The air-pumps of vertical engines are almost invariably arranged vertically, and driven from the cross-head by levers at about half speed of piston; and so also are the great majority of those used for horizontal engines. All vertical air-pumps are made of the bucket type, with a valve in the bucket; a stationary valve called the foot-valve is placed beneath the bucket, and another called the delivery-valve is placed above the bucket. The barrel of an air-pump for a stationary engine is generally made of cast-iron.

This, however, should not be done if there is the least trace of acid or salt in the water, as cast-iron then soon becomes spongy and useless. Cast-iron possesses the merit of great durability, when not placed at a disadvantage on account of liability to corrosion. Brass is, however, successful in every ordinary case. The foot-box which contains the foot-valve, and upon which the barrel is fixed, may be made of cast-iron, as it is very largely protected from corrosive influences by the unbroken skin of the casting. The hot-well is of cast-iron, and should always be cast separate from the barrel, for facility of renewal. The foot-valve should be accessible for examination with a minimum of trouble and loss of time. One large whole valve may be adopted, or a series of smaller ones. The material of the valves may be of soft, thick india-rubber, vulcanized fibre of less thickness, flexible brass, or light, solid brass castings, fitted loosely upon fixed spindles. Large valves of cast-brass are now seldom employed, owing to the knock at moderately high speeds, which is almost inseparable from the use of a single valve, lifting so far as to give a free delivery. Soft rubber is acted upon very largely by excess of mineral oil, so that its application is not always to be recommended. In each case a guard is required to regulate the lift of the valve, which in all cases is greater than that of the valves of feed-pumps. Springs are also used with advantage, to accelerate the closing of the valves, and to promote easy and quiet working.

Velocity through foot-valve. — The velocity to be allowed through the foot-valve is limited by the necessity for the maintenance of a good vacuum in the condenser. A perfect or absolute vacuum is to be desired, but if this were possible of attainment the condensed water could only be impelled through the foot-valve by reason of a head of accumulated water

behind it, which would obviously bar the passage of air and vapour, so that the vacuum must suffer. If it were necessary and possible to pass the condensed water through the foot-valve with a constant velocity of 7 feet per second, while an absolute vacuum existed in the condenser, the water in the condenser would necessarily stand at a level of from 2 to 3 feet above the foot-valve, varying with the arrangement of valve. This would cause a loss of 7 to 10 per cent. in the vacuum. By increasing the area of foot-valve so as to reduce the velocity to 4 feet per second, the loss of vacuum may be reduced to about 3 per cent., and water would stand at a level of about one foot above the foot-valve. The water does not actually stand above the foot-valve, but an equal force is supplied by the tension of the remaining vapour and the liberated air in the condenser, so that the effect upon the vacuum is the same. The area of opening through the foot-valve should suffice to pass the condensed water, the condensing water of forty times the weight, and an equal volume of air, or eighty times the volume of the condensed water, without the mean velocity during the stroke exceeding 4 feet per second. In some cases the velocity reaches 7 feet per second, and in exceptional cases may be allowed to reach 10 feet per second, but this must be understood to involve a sacrifice of vacuum. The velocities obtained from the volume swept by the air-pump bucket are in each case somewhat greater.

Foot-box.—The foot-box is of cast-iron, arranged to receive the mixture of water and air freely from the condenser, and to allow the whole to pass to the air-pump without possibility of accumulation of air, either beneath the foot-valve cover or around the bottom end of the barrel of the air-pump. The bottom of the foot-box should be arranged to lead with a moderate slope

to the foot-valve. The foot-valve should be arranged to oppose the least possible resistance to the flow of water and air, should be of light weight, and yet return promptly, so as to retain the charge, otherwise the vacuum will suffer. The opening should be free from ledges, shoulders, and abrupt changes of section to oppose the current, which is of little, if any, less importance than ample sectional area. For the same reason the cross-bars in the foot-valve seating, which are necessary with many kinds of valves, should be as narrow and smooth as possible; they should also be cut to a thin edge facing the current.

Bucket- and delivery-valves.—The area of opening through the bucket- and delivery-valves should be not less than through the foot-valve, if convenient. This is, however, not imperative, as a deficiency here would not seriously affect the vacuum, though it would cause the imposition of increased work upon the air-pump. A circular foot-valve, or a series of smaller valves, is sometimes fitted in the bottom of the air-pump. When well designed, this is an excellent arrangement. Its weakest feature is the inconvenient access to the foot-valve. In some cases this is only reached from above, after the removal of the delivery-valve and the bucket or the air-pump bucket itself. In other cases the valve is reached by a side door, or from below, when it is necessary to elevate the foot-box to give sufficient space. In connection with a foot-valve of this type, the existence of cavities in which air may collect is of no consequence, as it is cut off from the air-pump by the foot-valve, when the latter is closed during the descent of the bucket. But all irregularities in the surface or the form of sectional area by which the current may be impeded, or in the direction of the latter, should be minimized. The slip or loss of water at the valves

is obtained as already described in connection with feed-pumps. When single valves are employed, a greater lift is necessary than when a series of smaller valves are adopted.

Air-pump bucket.—The air-pump bucket is a plain piston with passages through it, covered by the bucket-valve. Under favourable conditions it may be made of cast-iron, with or without the edges covered with brass. In most cases, however, a solid brass casting is to be preferred. The edge is plain, packing being unnecessary, and leading to accidents, by jamming in the barrel when worn thin, or when the bucket is allowed to drop sufficiently for the rings to open, and hook against the bottom of the barrel, as the latter is seldom counterbored to a taper. When the foot-valve is fitted in the bottom of the air-pump barrel, the bucket cannot descend so far. Sometimes grooves are turned in the edge of the bucket, but they are not really necessary. The air-pumps of marine engines are often packed by rope or hard yarn, coiled in a wide groove and projecting slightly beyond the edge of the bucket, so as to press against the bore of the barrel. A little clearance round the edge of the bucket is not very objectionable, as the water left between the delivery-valve and the bucket has an opportunity to drain back into the bottom of the barrel, during the first part of the return stroke. The bucket-valve then rises more freely, to allow the air to pass through it as the bucket descends, which improves the working of the pump, but has no effect upon the vacuum. The bucket need not be very deep, and should work as close to the bottom as possible, so as to leave the least possible amount of water in the pump for the next charge of air to rise through. The bucket is often fitted on a coned rod, with the large end down, and a small

shoulder provided in addition. The valve-guard is placed upon the rod above the bucket, and the cotter applied against the guard, driven home, and secured by a pin. In other cases, and perhaps the majority, the rod is screwed into the bucket and riveted secure, the guard being afterwards secured by a nut screwed down upon it. The rod, in all cases where exposed to corrosive influence, should be of muntz metal, phosphor bronze, delta metal, or equivalent material. In horizontal air-pumps the piston should be made to a free fit, without appreciable shake.

Hot-well.—The hot-well should be made of diameter sufficiently large to provide ample space around the delivery-valve, or each one of the series of delivery-valves, as may be adopted. Usually the diameter of the hot-well very much exceeds that of the air-pump barrel, so that the top end of the barrel should be tapered to suit; this should be carefully proportioned with a view to minimize the space intervening between the bucket- and the delivery-valves, without unduly encroaching upon the approach of the water to the valves. The discharge-branch from the hot-well should be so arranged as to leave about two inches of water above the delivery-valves, so as to prevent them from drawing air. If more water is left upon the valves, the discharge of the load brought up by the ascending bucket is correspondingly and unnecessarily obstructed. The feed-pump supply-pipe, which draws water from the hot-well, should also be at all times covered with water.

Horizontal air-pumps.—Horizontal air-pumps are often adopted to stand above the floor of the engine-room, when it is not convenient to place them below the floor. They are usually driven by a backward extension of the piston-rod, and therefore at the same speed as the

piston. They are made either single-acting or double-acting. A chamber is filled with water, which is set in motion by means of a piston in the double-acting class, or a ram in the single-acting class. The pulsating upper surface of the water causes the indraft from the condenser and the discharge into the hot-well. The condenser and the hot-well are generally placed above the air-pump for efficiency and convenience, so that one set of valves opens downwards and the other upwards. Both sets of valves should be adequately loaded, to close clearly in advance of the water, for the reasons already given. By the use of the water surface to draw and force the air and water in this way unnecessary resistance, by reason of the necessity for air to rise against a head of water, is avoided. In high-speed engines the proportions and arrangements adopted must be such as to ensure uninterrupted contact between the water and the piston or ram. In the latter case the pointed end is of great assistance. The Allen single-acting air-pump is of this description, and has been used with success, though usually the capacity of the condenser is very small. Illustrated accounts of these engines were given in *Engineering*, vol. v., pp. 88, 119, 143, 159, 184, and 200. In these air-pumps the ram should be made sufficiently light to resist bending when overhanging the stuffing-box in the pump, and when withdrawn. In the latter case the weight is, however, fairly well supported by the piston-rod. Disregarding the effect of the additional weight in the point, the ram, if made in iron, would be on the point of floating, if its thickness equals $\frac{1}{30}$th part of its diameter, or about $\frac{1}{34}$th part if made in brass; therefore these proportions should be approached as nearly as may be practicable. An ordinary stuffing-box would be likely to draw air, and give considerable trouble, unless kept

in the most perfect condition. But by the provision of a lantern supplied with water from the hot-well, all inward leakage of air is avoided, and also all necessity for keeping the gland very tight.

Speed of air-pumps.—Though it is quite possible to run air-pumps at a high speed with perfect success, yet the indiscriminate adoption of a high speed cannot be recommended, as there is more wear and tear upon the valves, and the vacuum suffers. If the speed of a bucket of a vertical air-pump rises very high, it becomes difficult to provide sufficient area of opening through the bucket for the passage of the water at a reasonable velocity. It is therefore better to avoid a speed exceeding 80 revolutions per minute.

Slow driving of pumps.—When an engine works at a speed too high for the pump, the latter may be separated and driven from the main engine by means of ropes or belts, at any required speed. This arrangement is also convenient in allowing a wide choice of positions for the air-pump. But freedom of drainage from the condenser to the air-pump is in all cases a necessary condition to be observed. Direct driving of the pumps, by one or more steam cylinders, exclusively devoted to the purpose, is also often successfully adopted. This is especially convenient in allowing adjustment of speed to suit variations in power, or in condition of condensing water.

Capacity of air-pump.—The capacity of the air-pump of a jet condensing engine should be sufficiently great to pass the maximum quantity of water which may at any time be required for injection into the condenser, as well as the condensed water, and all air which may be introduced in the feed-water to the boiler, or the injection water to the condenser, or derived from other sources (as explained in connection with the feed-pump),

or by leakage in the exhaust-pipe or connections, or at the stuffing-boxes, or the drain-cocks of the low-pressure cylinder. When all details are properly arranged, and kept in good condition, the quantity of air should not exceed one-third of the volume of water discharged, at an absolute pressure of 3 pounds per square inch. In connection with the design of the pump and valves, the air may, however, be assumed to equal the volume of water. The volume of water required depends upon the temperature at which the injection water is received, and that at which the mixed water is discharged. When the difference between these temperatures is 34° F., the weight of injection water required is thirty times as great as that of the live steam to be condensed. When the difference in temperature is 26°, the weight of injection water will be forty times that of the steam. In the very exceptional case in which the difference in temperature is 17°, the weight of injection water will be sixty times greater than that of the live steam to be condensed. The minimum capacity for the air-pump to work efficiently should be therefore equal to the passage of liquid water equal to 120 times the weight of steam to be condensed. In almost all cases the pump is made of greater capacity than this; and in all cases allowance should be made as already explained in connection with feed-pumps, on account of loss of water at the valves. In any case the capacity of the air-pump should be estimated directly or indirectly upon the quantity of steam used, and not upon the horse-power. By estimation upon horse-power, the air-pump of an engine which works upon a low consumption of steam fails to receive due credit. The capacity is however usually, for convenience, based upon the volume of the low-pressure cylinder. If steam is expanded in this cylinder to an absolute pressure of 7 pounds per square

inch, the lowest admissible capacity of the air-pump is $\frac{1}{24}$th part of the capacity of the cylinder, including clearance, and this is occasionally adopted for air-pumps which are intended for ordinary use in connection with surface condensers, arranged for occasional use of jet condensation. Much more frequently, however, the ratio is one-eighth or one-tenth. The former ratio is only possible under the most favourable conditions, when two single-acting or one thoroughly efficient double-acting pump is adopted, by which a more uniform discharge of water and air from the condenser is effected than is possible with one single-acting pump. With one single-acting pump, excellent results are obtained, when the ratio adopted is one-twelfth. A small pump is less costly to provide, works more steadily, and is more easily driven, but fails to produce a good vacuum. When a pump of large capacity is adopted, the velocity of water through the valves is usually low, whereby the resistance is minimized. This may possibly be the chief cause of improved vacuum arising from the use of a large pump. The ratio should be estimated upon the total calculated discharge of the air-pump per minute, compared with the volume of the low-pressure cylinder multiplied by the number of single strokes per minute. This will cover every case of single or double-acting pumps, whether directly driven at the same number of revolutions as the engine, or separate, as when driven by rope or independent steam-engine.

Circulating pump.—In a surface condensing engine a circulating pump is necessary to pass the cold water through the tubes of the condenser. This is often made precisely similar to the air-pump, so that if anything goes wrong with the bucket of the latter the circulating pump bucket may be put into the air-pump and work continued by means of the jet injection. In

this case the diameters are identical, but the stroke may be varied, as the capacity of the circulating pump need not exceed one-half or two-thirds that of the air-pump, as in the circulating pump it is not necessary to provide a margin on account of air given off by the water. When no feed-heater or economizer is used, a circulating pump of excessive proportions leads to excessive cooling of the condensed water, beyond what is necessary on account of the vacuum. On this account such a circulating pump is sometimes worked with the intake valve partially closed, so that the condensed water is obtained from the hot-well at a higher temperature. This, however, causes irregular working, and increased wear and tear in the circulating pump. The necessity for it should therefore be avoided by the provision of a bye-pass, to admit water from the discharge-pipe to the suction-pipe under control of a valve. A double-acting circulating pump is of the differential type, in which water is drawn in during the upward stroke and discharged equally during both upward and downward strokes. A bucket and foot-valve are required similar to those of a single-acting pump. The pump-rod must pass through a stuffing-box, and be enlarged to a diameter = diameter of bucket × ·7 in such stuffing-box. By this means the area of the rod is one-half that of the bucket, and the discharge equally divided. The delivery-valve is unnecessary, but is sometimes added. Sometimes the bucket, with openings through it, is replaced by a solid piston, when a separate passage is required to communicate between the top and bottom of the air-pump barrel. In some cases this is cast in the pump barrel, after the manner of a steam-port in a locomotive cylinder. This is a neat arrangement, but is apt to lead to contracted passages and very difficult and risky joints. It should

therefore be discouraged and separate pipes adopted. Two valves are still required. The object with which a differential pump is adopted for this purpose is to render the flow more uniform through the tubes. Obviously this object would not be secured if the suction side of the pump were connected to the condenser. Therefore a differential pump must be arranged to force water through the condenser. This is generally the most convenient course to adopt, as the water may be admitted at the bottom of the condenser, and rise in opposition to the course of the steam. Thus the hottest water meets the hottest steam, and a better vacuum is secured than is possible when the hottest circulating water is in contact with the steam at the bottom of the condenser. For special reasons the water may be drawn through the tubes, but it should always leave the condenser where the steam enters it. An air-vessel should be provided between the circulating-pump and the condenser, to equalize the flow and prevent shock upon the tubes. This is almost equally useful, also, when the water is drawn through the tubes. The bucket of the circulating pump is generally packed with hard yarn coiled tightly around it. This is, however, unnecessary. Plain grooves, turned in the edge of the bucket, oppose all necessary resistance to prevent serious leakage. The diameter of the bucket should, however, be such as to give only a free fit without such clearance as is admissible in an air-pump.

Centrifugal pumps and piping.—Centrifugal pumps are sometimes adopted as circulating pumps, for which purpose they are most efficient, performing their work with absolute uniformity, and singular absence of shock or wear and tear. Centrifugal pumps of small sizes are sometimes driven by belt from the main engine. But much more frequently they are driven by a special

engine combined upon the same bed-plate, the crank-shaft and the pump-spindle being in the same line, connected together. Both types are used for lifting water to a cooling-pond or stack, and for other purposes temporary or permanent. In connection with the use of such pumps, the chief precautions to be observed are the avoidance of any pockets of air in the pump-casing or the pipes; of inward leakage of air at joints and the pump-spindle stuffing-boxes; and of resistances arising from irregularity of form or section in any part through which the water is required to flow. If the whole length, from the suction end to the pump-casing, is under pressure, cocks may be applied for the discharge of air from any pockets which cannot conveniently be avoided, such cocks to be opened before starting the pump. Similar cocks on the discharge-pipe may be opened at any time, but are seldom absolutely necessary. Accumulations of air in centrifugal pumps exclude water; the revolving disc is unable to act effectively upon air; therefore the pump will not generate a current when largely filled with air. Accumulations of air in pipes may be swept away by a brisk current. But if such air should fail to be swept away, it blocks the pipe almost as effectively as a piece of wood, with slight relief by reason of elasticity. Pump pipes should therefore be placed so that air will be carried forward by the current, *i. e.* on a rising gradient. Inward leakage of air is seldom so extensive as to interfere with the work of the pump if the pipes are laid on a rise, but it may cause great trouble in starting the pump. If the suction end of the pipe is fitted with a foot-valve, the entire pipe may be filled with water to a level above the pump-spindle, when any leakage will be at once located. Many such pumps are, however, unprovided with foot-valves, but are provided with

some kind of head-valve or door, and with a steam ejector for exhausting the air from the pipes. With care, a leakage may be then located by wetting the hand and applying to all joints while the pipe is exhausted by the ejector. A gurgling sound will then be heard, though it may be necessary to stop the ejector for the purpose. When a belt is used for a centrifugal pump it should receive special attention as to strength and uniformity of joints; also as to protection from the weather.

Steam-pumps.—Separate pumps, each driven by its own engine, are used in very small sizes for boiler feeding. In larger sizes they are used for general pumping, boiler washing, and fire extinction. In all sizes such pumps of the duplex type work well, though usually extravagant in steam consumption. In the duplex type, two complete engines are used, placed side by side, usually horizontal. The valve motion of each is controlled by the piston-rod of the other one. Other pumps are generally vertical, with either one or two cranks. In some, each pump is double-acting, consisting of a bored cylinder, and possessing four valves, of which one suction- and one discharge-valve communicate with each end of the cylinder. In other pumps rams are used in a similar manner. Small steam-pumps, however, work with great perfection when single-acting, by means of a single plunger, the steam slide-valve being adjusted to give more steam at one end than the other. A pump of this or the duplex type should be used for boiler feeding in all cases, as an alternative to the engine feed-pump, for occasional use. The capacity should be such as to require a moderate speed for the discharge of ordinary work. A pump for fire extinction should discharge from 250 to 1000 gallons per minute, and should be regularly used for brigade

practice or for boiler washing, in order to ensure that it be kept in order. Such a pump is usually supplied with steam from the main boilers by means of special pipes, and should be so placed and arranged as to incur little risk of interruption by broken steam- or water-pipes, or other accidents, before the boilers themselves are disabled. Fire-pumps are sometimes proportioned to work with steam of reduced pressure, by which means their usefulness is greatly increased. This measure is perfectly safe, if efficient relief-valves are provided, which step should never be omitted with force-pumps of any kind. The cost of a relief-valve is very trifling, and experience shows it to be absolutely essential to safety. Neighbouring works sometimes have their pipes connected for mutual assistance in case of fire, and to increase the amount of water which can be brought to bear upon any given point. Full area stop- or sluice-valves are necessary for cutting out various portions of pipe which may become damaged, so that work may continue with the remainder.

Precaution against frost.—During winter, all pumps, pipes, and vessels containing water are liable to freeze, and, especially if cast-iron enters into their construction, should be drawn off quite dry when left out of use, even for a comparatively short time, otherwise rents will occur and stoppage of work will follow, the loss involved in which may be much greater than the direct expenditure upon the replacement or repair of the damaged parts.

CHAPTER XXXIII.

GOVERNORS.

Purpose of governor.—A governor is required in connection with each stationary rotative engine for the purpose of effecting an automatic adjustment of the steam supply in accordance with variations in the load and fluctuations in the steam-pressure, so that the speed is maintained uniform. In almost every case, the effect of centrifugal force upon rotating bodies of considerable mass is utilized.

Watt's governor.—The ordinary governor, invented by Watt, consists of two revolving pendulums, each of which is suspended at its upper end near to, or upon the revolving spindle. At the lower end of each pendulum is a heavy ball, which naturally occupies a certain position relatively to the spindle at a certain speed. This position is such that the vertical distance from the line joining the centres of the two balls to the point x in Fig. 84, where the centres of the two arms intersect with the centre line of the spindle, in inches $=$ $h = \left(\frac{188\cdot2}{\text{number of revolutions per minute}}\right)^2$. The point x need not coincide with the pins of the arms, and in fact may not be within the arms at all. At the speed thus defined,

the weight of each ball acting vertically, and the centrifugal force acting radially outwards, are in exact equilibrium with the pull upon each arm. If, however, the speed of revolution should increase, the centrifugal force will increase, and the balls, while still revolving, will assume a higher position, as dotted in Fig. 84, in which the new height of cone h corresponds to the new speed of revolution. The converse action ensues upon a

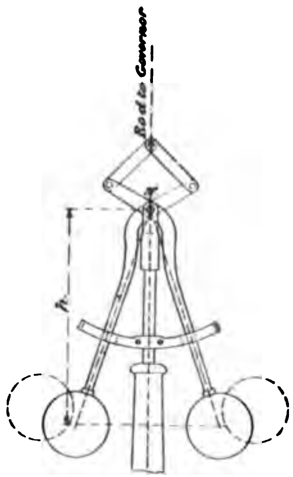

Fig. 84.—Watt's governor.

reduction in speed. Therefore each position of the balls corresponds to one particular speed of revolution. When a direct connection is made between the governor and the steam adjusting gear, it follows that one particular phase of steam regulation corresponds to one particular speed. Therefore in the simplest case full power will be found to be developed at only one particular speed, and lower powers at correspondingly increased speeds. In practice this discrepancy is

approximately met by variations manually effected in the lengths of the rods connecting the governor to the steam gear. For this purpose one or more couplings with right- and left-handed screws and hand lock-nuts are conveniently provided. Usually the highest position of the balls, and therefore the highest speed, corresponds to the least supply of steam which will just move the engine; and the full steam supply is secured when the balls are in their lowest working position, and therefore moving at a considerably reduced speed. The adjustable coupling should be arranged to work within limits which must be fixed to avoid all possibility of interference with the action of the governor. In other words, the whole system must effect the admission of more and of less steam to correspond with the power required, whether the coupled rods are set at their greatest or smallest length. In all cases the rods, pins, levers, and connections, including stuffing-boxes, valves, &c., are subject to frictional resistances, which interfere with freedom of motion, and which can only be overcome by the application of definite force. The force must be supplied by the governor, and the variation in speed—whether of increase or decrease—must be sufficiently extensive to produce a corresponding positive or negative change in the centrifugal force. In this way the variation in speed becomes very much greater than would be anticipated, and certainly much greater than is desirable. Every attention should therefore be given to the design and the maintenance in clean, free condition of the whole, so as to ensure the reduction of such resistances to their lowest limits. In some engines the governor spindle is bored and a rod passed down to the lower slide of the governor. This is an excellent arrangement, but particularly liable to the defect in question. In some cases the governor is so arranged

that it does not directly actuate the regulating motion of the engine, but initiates such operation, while the chief part of the work is performed by steam, or by the ordinary motion of the engine.

Isochronous governor.—It is possible to arrange that the cone of revolution shall remain uniform in height, however far the balls may diverge or converge. When this is the case, the balls will remain in any position in their path, so long as the speed of revolution remains normal. If, however, the speed should suffer a slight increase the balls will diverge, and continue to do so until the normal speed is resumed, or the balls reach the extremities of their path. Upon the resumption of the normal speed the balls remain where they happen to be, until the speed falls below normal, when an opposite action is developed. This condition is absolutely secured when the balls follow a parabolic curve of suitable proportions, and the third force acts upon each ball in a direction perpendicular to the tangent of the curve at the point of contact. But the means whereby this condition is secured are often attended by a much greater amount of friction and complication than those whereby the balls are guided in a circular path. As in all other governors, the effect of friction is to delay action, and to increase the interval between the upper and lower limit of speed within which the governor will effect the regulation of the engine. The balls may be suspended by springs, whereby the amount of friction is reduced, but except for small governors this is not a safe plan, having regard to the stresses to which such springs would be subjected.

Governor approximately isochronous.—The balls may, however, be suspended by pins and constrained to move in circular paths approximately equivalent to parabolic ones. With this object the point of suspension of each

arm is placed on the side of the spindle opposite to the ball, so that the two arms cross each other against the spindle, as in Fig. 85. By this means the range of the governor may be increased largely under very small variations of speed. The chief faults of the Watt governor are therefore largely avoided. But the inertia of the balls is apt to cause the regulating motion to be continued beyond the point at which the correct adjustment would be obtained. When this is the case the speed oscillates within irregular limits above and below

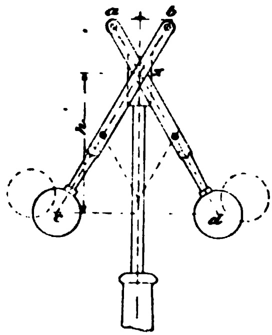

Fig. 85.—Governor with crossed arms.

the normal speed, or the governor is liable to allow "hunting." Such oscillations are more pronounced in the absence of excessive frictional resistance. The period of revolution is that due to h, the height of cone, as in Fig. 85. Hunting may be avoided by the adoption of such proportions in the governor as to give a reduction in the height of the cone of revolution as the balls rise. In some cases an opposite course is inadvertently adopted, and the height of cone increases as the balls rise, when the governor utterly fails to perform efficient duty. When the distance from centre to centre of pins a and b in Fig. 85 is one-quarter of the length from the

centre of the pin b to the centre of the ball c, the height of the cone decreases very slightly, as the balls rise from the position in which the arms make an angle of 30° with the spindle. If the distance $a\,b$ is one-third of the distance $b\,c$, the arms should not be allowed to fall below 33°. In either case the sensitiveness of the governor is increased fourfold above that of such a governor as shown in Fig. 84, and the tendency to oscillation is practically avoided.

Common governor.—In most cases each governor arm

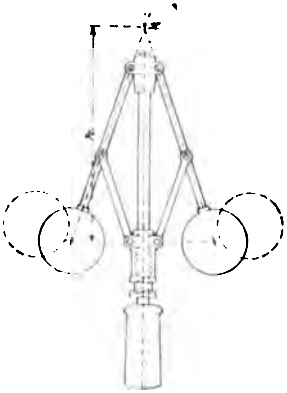

Fig. 86.—Common arrangement of governor.

is secured by a pin on the same side of the vertical spindle as the ball, as in Fig. 86. The effect of this upon the sensitiveness of the governor is of a nature opposite to that due to crossed arms, so that in comparison with a governor of the usual type, one with crossed arms possesses still more advantage.

Loaded governor.—In many classes of governors the total weight is divided. The two balls remain practically counterparts, on a reduced scale, of those shown in Fig. 84, acted upon by gravity and centrifugal force,

while the larger portion of the weight forms a central load, fitted to slide upon the spindle, and arranged to remain unaffected by centrifugal force, but by its weight to oppose the lifting of the balls due to their own centrifugal force. When arranged as in Fig. 87, the central load rises at the same rate as the balls measured in a vertical direction. According to the principles of virtual velocities and the parallelogram of forces, the whole of the weight acts in opposition to centrifugal force, precisely as if it existed only in the two balls. But as only the balls are exposed to centrifugal force,

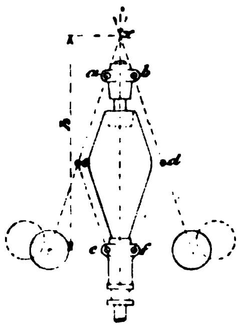

Fig. 87.—Loaded governor.

and as this centrifugal force is held in equilibrium by the same weight as before, it follows that the necessary amount of centrifugal force must be due to a speed greater than that necessary in the original case of simple balls. In such a case the cone, as measured by the arms and the base joining the centres of the two balls, is the apparent cone of revolution. The virtual height of the cone of revolution is found by dividing the apparent height by the weight ratio of the governor, which is found by dividing the sum of the weights of the central load and the two balls by the separate

weight of the balls. This applies to all governors, however arranged, in which the separate weight rises vertically at the same rate as the balls.

Ordinary arrangement of loaded governor. — The majority of governors in which a separate weight is adopted, or "loaded governors," are constructed with the revolving weights mounted at points *c* and *d* in Fig. 87, by which means the central load is caused to rise at double the vertical rise of the balls, by which means the virtual weight ratio of the governor is also doubled.

Variation in speed must suffice for centrifugal force to overcome frictional resistance in system.—It has already been stated that the frictional resistance which is exercised by the system, in opposition to adjustment by the governor, must be overcome by a positive or negative change in the centrifugal force, due to a change in speed. The approximate amount of resistance can be obtained in ordinary cases by the use of a spring balance applied to the governor at rest. Before any movement of adjustment can be completed, the speed must increase so as to develop an excess of centrifugal force sufficient to overcome the frictional resistance, and usually, in addition, to allow for such increase in the speed as may correspond to the new position of the governor-balls at the end of the act of adjustment. The centrifugal force of the two balls, due to the two speeds and positions, may be calculated, being =

$$\cdot 00034 \times \text{weight in lbs.} \times \text{radius of revolution in ft.} \times (\text{no. of revolutions per min.})^2.$$

and the difference should be not less than equal to the amount of frictional resistance to be overcome. Conversely, when speed is reduced, an opposite action is required. In most cases, the negative amount may be assumed to be precisely equal to the positive one. As

an additional check upon each other the figures may, however, be calculated through. In some cases, the frictional resistance during work is largely increased by reason of the application of steam-pressure. In others, the frictional resistance of a cut-off gear will be found to be practically beyond the direct control of the governor, within the required limits of speed. But in all cases, calculations of the nature indicated will be found useful in showing the limits to which any particular method of treatment may be carried.

Comparison of loaded governors.—Applying the principles described to a series of governors of the type shown in Fig. 87, in each of which the combined weight of material used in the two balls and in the central load is constant, the governing power of each governor in the series is precisely equal to that of each of the others for equal percentage of variation in the speed, under the following conditions:—(1) The weight ratio of the governor varies as the $\frac{3}{2}$ power, or as the square root of the cube of the ratio of speed. (2) The length of arm, radius of revolution of centres of balls, and the heights of the apparent and virtual cones of revolution, vary inversely as the square root of the ratio of speed. Table XX. gives the leading particulars of a series of governors which answer to these conditions, beginning with one, in accordance with Fig. 84, for a speed of 33 revolutions per minute, and with balls of 180 pounds each. Each of the succeeding governors is in accordance with Fig. 87; and in each case the total weight of the two balls and the central load is 360 pounds. In each case also a difference of 1 per cent. in speed will cause an increase or decrease of about 2 pounds in the centrifugal force pertaining to each ball, when working at the radius given. The table also strictly applies to a series of governors with the revolving weights placed at points *c*

TABLE XX.—COMPARISON OF A SERIES OF LOADED GOVERNORS.

Speeds of governors. Revolutions per minute.	Weights in pounds.			Weight ratio of governor.	Height of cone (inches).		Radius of centre of balls.
	Ball.	Central load.	Ball.		Apparent.	Virtual.	
							inches.
33	180	0	180	1·00	32·52	32·52	18·00
66	63·6	232·8	63·6	2·83	22·9	8·13	12·72
99	34·6	290·8	34·6	5·20	18·6	3·61	10·39
132	22·5	315·0	22·5	8·00	16·2	2·03	9·00
198	12·2	335·6	12·2	14·70	13·5	·90	7·34
264	7·96	344·1	7·96	22·63	11·5	·51	6·36

and *d* in Fig. 87, with a suitable modification in the third and following columns. If the central load is so connected that it lifts twice as far in a vertical direction as the vertical rise of the balls, the weight of the central load in each case will be only one-half that given in the table, and similarly with other proportions. The virtual weight ratio thus =

$$\frac{\text{Weight of central load} + 2 \text{ balls}}{\text{Weight of 2 balls}} \times \frac{\text{Rise of central load}}{\text{Rise of each ball, measured vertically}}.$$

The force of 2 pounds just given as the difference in the centrifugal force of each ball, due to a change of 1 per cent. in speed, may be increased or diminished in its transmission to the point of application, just as its range is diminished or increased. Thus if each ball moves to a position 1 inch further from the spindle, or measured horizontally, while the rod is pulled or pushed $1\frac{1}{2}$ inches, the force upon the rod due to the action of the two balls will be 2×2 pounds $\times$ 1 inch $\div 1\frac{1}{2}$ inches $= 2{\cdot}67$ pounds. But if, under conditions otherwise equal, the rod moves only three-quarters of an inch, the force due to the two balls becomes 5·33 pounds. In other words, what is gained in force is lost in range, space, or velocity. The distance or range through which each of the series of governors exercises a certain force, by reason of a certain percentage of variation in the speed, varies in proportion to the length of arm, or in similar cases in proportion to the apparent height of cone.

Lightness of moving parts in loaded governor.—The greatest advantage of a loaded governor, as compared with one of the original type, is the comparative lightness of arms, pins, links, and other details, which lead to an important reduction in frictional resistance.

Loaded governor with crossed arms and links.—In

some cases the arms of a loaded governor are crossed, in the same way as those of the unloaded governor shown in Fig. 85. This is just as beneficial as in the unloaded case, provided that the arrangement is such that upon any change in position the load rises vertically, in exact proportion to the rise of the balls. In the more usual case, in which the central load rises twice as rapidly as the balls, the links should be crossed as well as the arms, which arrangement is shown in Fig. 88. The lengths $a\,d$, $b\,c$, $c\,f$, $d\,e$, are all equal; $a\,b$ also $= e\,f$.

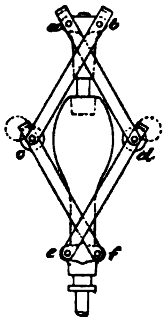

Fig. 88.—Loaded governor with arms and links crossed.

Speed of loaded governor.—The speed of revolution of a loaded governor may be obtained approximately in the manner above set forth. But in a loaded governor the weight of the arms and links possesses a greater influence upon the speed than is the case in an unloaded one. The number of revolutions per minute of a governor in which the central load rises twice as much as the balls, much more closely =

$$\frac{188{\cdot}2}{\sqrt{\dfrac{\text{Apparent height of cone of revolution in inches} \times \text{weight } (2 \text{ balls} + 1 \text{ arm} + 1 \text{ link})}{(2 \times \text{weight of central load}) + \text{weight of } 1\tfrac{1}{3} \text{ links} + \text{weight of } 1 \text{ ball.}}}}$$

The weights to be stated in pounds. The links are assumed to be of uniform section throughout.

Horizontal governors.—In many cases the governor spindle is laid horizontally, when the action of gravity upon revolving weights becomes inoperative. Springs are therefore necessary to cause the return of the balls to their low-speed position. In some governors, in which a vertical spindle is used, springs are adopted in conjunction with the weight of various parts. In all cases, however, revolving details of considerable weight are necessary for the development of the amount of centrifugal force requisite to furnish the motive power for regulation. Most of such cases are capable of analysis by the assistance of a spring balance. The centrifugal force of the revolving balls or blocks may be ascertained in precisely the same way as in vertical governors. The use of springs gives a facility for fine adjustment (without cutting off or adding weight), which in some cases is most convenient, and is thus resorted to even in cases in which the main action is by weight.

Fluid control of governor.—Many governors which, in use, have been found to be liable to oscillation, have been much improved by the addition of a dash-pot, or equivalent device. Usually a fluid is confined in a cylindrical chamber, and a piston arranged to work against the fluid, which is allowed to escape from one side to the other through a graduated orifice. This allows motion to take place slowly with freedom, but opposes great resistance to more rapid motion. Such a dash-pot may be placed in any convenient position. Messrs. Clayton and Shuttleworth place one inside the central weight of a loaded governor. Another arrangement for this purpose is Higginson's regulator, which consists of two vessels partially filled with mercury. These are attached to arms which are mounted upon the lever weigh-shaft of the governor, one in each direction, and connected below by a tube, which allows

the passage of mercury from one vessel to the other, and above by a corresponding air-tube. When one vessel is at a lower level than the other, it becomes filled with mercury from the upper one. If now the conditions change, and the loaded vessel is required to rise, it exercises a considerable steadying action upon the governor, and so prevents the formation of oscillations of very short period.

Governor actuating clutch gear.—In many cases, governors have been applied in such a manner as to effect their purpose by clutching into gear, alternately, one way or the other, a nest of three wheels, by which any amount of power is available for the essential act of regulation. This plan has been largely used for water-wheels, in the regulation of which large power is required at a slow speed. Modifications of the same principle have, however, been used for quick-running steam-engines, in some of which frictional contact gear or fine-toothed gear has been adopted.

Governor actuating steam-gear.—Regulation by steam-power possesses great advantages, as by its means the governor is only required to initiate the operation of regulation, but is relieved of the heavy work. Consequently *all* efficient governors will work within much closer limits of speed when applied in connection with regulation by steam-power. Fig. 89 shows a steam relieving-gear in which a piston is actuated by steam under control of a slide-valve. The adjustment is effected by means of two levers, *a b c* and *d e f*. The upper lever is centred upon a fixed point *c*, and is moved by a rod connected to point *a*, and under the control of the governor. The lower lever is movable throughout, the point *f* being connected to the piston-rod and moving with it. Assuming that the piston and point *f* are at rest, any movement imparted to

point *a* must be transmitted on a reduced scale to point *d* and the slide-valve, which is so proportioned that any appreciable amount of motion in either direction must cause the admission of steam to one side of the piston, and its release from the other side. This

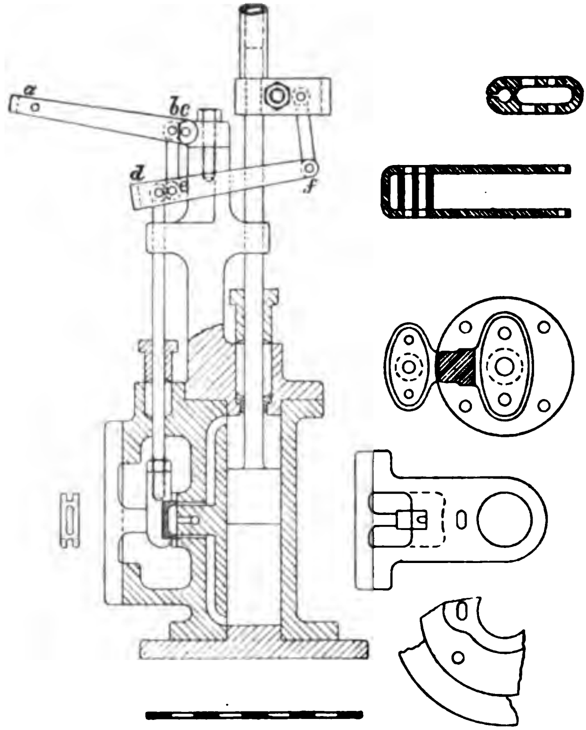

Fig. 89.—Steam relieving-gear for governor.

causes the piston to move, by which point *f* is moved and the valve restored to its central position, and the piston brought to rest. Consequently, the piston follows with great accuracy any motion imparted to point *a*. This may be either on the same scale, or it may be

increased or reduced, according to the proportions of the levers. The power of the piston depends upon its area and the pressure of steam applied to it. The degree of promptitude with which it responds to the movement imparted to point *a* depends upon the area of piston to be supplied, and the sufficiency of the ports for the passage of steam; in all cases the ports are of very small size. The amount of power required to be applied to point *a* depends upon the size of the slide-valve, the pressure of steam to which it is subjected, and the proportions of the levers, and must be clearly within the power of the governor to furnish, within the required limits of speed. Such a relieving-gear is well adapted for overcoming any steady resistance, and will usually give any required power, with only a very moderate amount of force applied to point *a*. If, however, the gear is required to oppose a force of variable amount, as in connection with some types of link expansion-gear, and especially if fluctuations suddenly arise, the steam in the cylinder will yield, so that no *rigid* control will exist. Such conditions will be met by the application of a well-proportioned dash-pot, or by the addition of a second cylinder similar to the first, but filled with a liquid admitted by a valve of the same description, but in an extreme case provided with guards to prevent it being forced off the face, to allow the escape of the fluid. A single cylinder may also be supplied with water instead of steam, to avoid yielding. One such cylinder supplied with steam may have its piston-rod attached to point *a* of a larger cylinder, which may be supplied with either steam or water, and so provide any amount of power or unyielding control. By such means the load imposed upon the governor is reduced to the absolute minimum, and the governor, being relieved from a large proportion of its total

resistance, is improved in sensitiveness and regularity. Such a gear may be applied to many other purposes, and in any position in which a small amount of power, manually or automatically applied over a short distance, is required to command a much larger power, acting through a limited but accurately proportionate distance. Lüde's gear, made by Messrs. Schäffer and Budenberg, is of this class.

Control of several cylinders.—Compound, triple, and quadruple expansion engines are imperfectly governed by means of the supply to the high-pressure cylinder only. Probably in few cases would any important gain be secured by connecting the governor to all the cylinders, on account of the quantity of gear required, and the resistances arising. But in exceptional cases, where this may be deemed desirable, the only means whereby to secure a successful result, will probably depend upon a steam relieving-gear.

Design, workmanship, and balance.—A governor should be throughout constructed with the greatest care, so that it shall perform its work steadily and with the least amount of friction. The spindle is a most important detail. It should be accurately turned all over, and this accuracy should extend to the bore, in case such provision is necessary. Its truth should be tested by careful comparison with a straight-edge, and by rolling over smooth level strips. The movable weight or slide of the governor should always be made of ample length, and if one-third of the length in the centre is chambered to a rather larger diameter it will work more freely. This chambering should, however, not be cast, on account of the probable effect upon the balance. The sliding surfaces may be made of cast-iron. The arms and links should be made of wrought-iron, or of gun-metal, and never of cast-iron. The pins are best

of wrought-iron, case-hardened and ground. In governor work pins are usually employed to connect two jaws or plates to one central knuckle, in which case the durability of the arrangement is increased by making the pin fit tightly in the knuckle, and preventing all motion there by means of a taper pin driven into a rimered hole. All movement and wear then take place at the outer surfaces, where, owing to the greater total width, the pressure, friction, and wear are less than in the knuckle. In a great measure the whole is thus prevented from getting into a loose condition. In some governors, each link is made of two flat bars or plates, for the sake of economy in manufacture. In its simple form this plan is quite destitute of side stiffness, and is unsatisfactory. The two parts may, however, be connected together by means of distance-pieces; and in some cases one flat bar may be bent round to form a very satisfactory arrangement. The whole governor should be tested for balance in several positions, when the parts are assembled together.

Freedom from strain in joints.—In an unloaded governor, stops should be provided for the balls to rest against when standing. In many cases, the bottom of the slide takes a bearing on the top of the column; and in all loaded governors the central load should bear in this way. When any of the weight is allowed to jam or bear on the jaws of the arms or links, or on the rods, levers, and connections to the regulating gear, damage is likely to be caused.

Automatic closing of throttle-valve.—In every case it is most important that means be provided to ensure the closing of the throttle-valve, when, from any cause, the governor comes to rest. Many serious break-downs have been caused by the want of such an arrangement.

Moscrop's speed and steam-pressure recorder.—Mos-

crop's recorder consists essentially of a clock to drive a strip of paper at a uniform rate; a small governor of great sensitiveness, whose sole purpose is to give a continuous record upon the moving paper of the speed at which the engine is moving; and a pressure-gauge, also arranged to give a continuous record upon the same paper of the pressure existing in the steam supply-pipe. When the speed of the engine is absolutely uniform, the recording governor gives a fine horizontal mark. Frequent variations in speed will cause rises and falls which may run into each other, so that the record appears as a band of considerable width. In such a case the main governor may possibly be improved by the addition of a regulator or dash-pot. But if the speed-record line follows the fluctuations of steam-pressure as recorded by the pressure-gauge, the fault is in the main governor, and is not to be affected by the addition of a regulator, which can only exercise influence over the minor variations, which are indicated by what may be called a wide line.

Driving of governors.—Almost all governors are driven by means of a pair of bevel-wheels in the base. In different cases the cross-shaft is driven by shafting and wheels, or by a belt, or by one or two ropes. The first gives the most absolute constancy of speed transmission; the second is almost as good, unless so far neglected as to allow great slip; but the third may be uncertain if the grooves are so sharp or the rope so soft as to cause variation in the depth to which the rope may sink in the groove. Belt or rope driving are more convenient for application under difficult conditions. Some risk of stoppage by the running off or breakage of belts or ropes is incurred, but if the governor is properly fitted with a stop motion, this causes little inconvenience and no danger. Pulleys for either belts

or ropes give great facility for making adjustments for various speeds.

Connection with gear admitting steam to engine.—In some cases the spindle of a governor is bored to receive a rod for the transmission of the motion from the governor to the regulating gear. In some other cases the motion is taken from the head of the governor, in the manner originally adopted by Watt. In most cases, however, a lever is used to take the motion from the lower end of the lower slide. In almost all cases levers are used in different parts of the connection. In some cases rods are used to transmit the motion by longitudinal movement; in other cases, shafts are used to act by torsional strength and stiffness. Rods should be allowed ample travel, so that they may be subjected to low stress. Of two alternative rods attached to levers of different length, but allowing equal travel in each case, the rod attached to the shorter lever will apply a smaller twisting stress upon the shaft. Consequently for equal travel the levers should be kept short, but for equal angles of rotation the levers should be kept long. By keeping the levers short in the former case, the diameter of shaft, its weight, and consequent friction may be kept low. As a rule, light rods should be kept under a continuous moderate tension. In some cases strong wires may be used, in connection with levers or grooved pulleys.

Steam-regulating gear.—In current practice on a large scale, the speed is almost invariably regulated by varying the cut-off. But occasionally in large engines, and frequently in small engines, a throttle-valve is used, sometimes the old disc-valve swivelling across the steam-pipe. But a more efficient arrangement is the equilibrium-valve, having an upper and a lower seating, each with a corresponding conical valve, the steam being admitted either between the two or outside. The

diameter of the lower valve should just pass through the upper seating, for facility in placing together, and for avoiding the occasion for inaccuracy which arises with separate valves. If this is judiciously effected, the equilibrium of pressure is scarcely disturbed. The use of a throttle- or equilibrium-valve of any kind is, however, opposed to economical working.

Emergency stop-gear.—In all works the desirability of a ready means of stopping the shafting in case of accident has been long felt. Clutches and arrangements for throwing separate shafts out of gear have been used, but not largely. During recent years electrical fittings have been much adopted for this purpose. In some cases wires are laid from various points to a valve in the engine-room, so that steam may be promptly shut off in case of accident. In other cases only a signal is made to the engineer in charge of the engines. In some cases a wire connection is made to a throttle-valve, or to the engine stop-valve, the particular valve being arranged to trip for the purpose. These measures are constantly found to be the means of saving lives and property, and are seldom abused. It is, however, necessary that means should be available to show the place from whence a signal to stop was sent.

References.—Mr. C. F. Budenberg read a paper in April 1891 before the Manchester Association of Engineers, which gives an excellent account of the governors of Buss and Proell, also the four-ball governor, the relieving gear of Lüde, and the expansion-valve of Schäffer and Budenberg, which cuts off the steam behind the valve-chest. Three papers, giving advanced treatment of several governors, by Professor Dwelshauvers-Dery, are translated and published in the *Minutes of Proceedings of the Institution of Civil Engineers*, vol. xciv., p. 210; vol. civ., p. 196; and vol. cx., p. 276.

CHAPTER XXXIV.

PIPES.

Loss of pressure in pipes.—Steam or other fluid passes along a pipe with uniform velocity, by reason of a difference in pressure at the two ends. The amount of this difference in pressure must be sufficient to overcome the resistances encountered in the pipe. When a pipe is used to draw fluid from a vessel in which it is comparatively in a state of rest, a certain amount of pressure is usefully consumed in the operation of producing motion. The amount of pressure thus accounted for is measured by the distance through which a body would fall freely under the action of gravity, in order to acquire the given velocity, and is equal in pounds per square inch to the weight of a column of the fluid of given height and one square inch in area. The actual pressure in the passage in pounds per square inch would then be obtained by subtracting from the original pressure the amount thus lost. But the amount of loss thus defined is based upon the assumption that the required velocity is obtained without incurring any loss, by reason of the formation of eddies or irregularities of current. In practice, such additional losses are always incurred, and the total loss is obtained by dividing the

loss first obtained by a co-efficient, which in different cases ranges between ·6 and ·9. Losses incurred in the generation of high velocity may be partially recovered in any subsequent reduction of velocity by the exercise of great care in the design of the passages. Such recovery cannot, however, in any case be more than partial, hence the area of all passages should be made as uniform as possible.

Effects of variation in diameter of pipe.—The resistances encountered in the pipe itself are much greater when the velocity is high. Consequently, as regards this point, the diameter of pipe should be as great as possible. But all steam-pipes, however well covered, are subject to loss of heat by radiation and conduction, which losses are greater in large pipes than small ones, and a point exists at which any advantage to be secured by increase in diameter, in minimizing the resistances, is neutralized by the loss by cooling. Pipes of large diameter are more costly than those of small diameter, on account of the greater circumference and thickness of material necessary to resist the pressure, so that the capital value of such pipes increases in approximate accordance with any reduction effected in the velocity of current.

Judicious velocities of steam in pipes.—Steam-pipes are found to give good results when the mean velocities through the pipes are about in accordance with Table XXI.

At low pressures the disadvantages attached to pipes of large diameter are less pronounced, and therefore somewhat lower velocities than those given in the table may be adopted. The quantity of steam is to be taken as dry steam, according to the weight of feed-water supplied, and to the pressure which may exist at any time at the part in question, making no allowance for

partial condensation. These velocities are a little lower than those given elsewhere as applicable to steam-ports. When steam is cut off at an early part of the stroke and only a small valve-chest is used, some allowance should be made in the area of pipes.

TABLE XXI.—APPROPRIATE VELOCITIES OF STEAM THROUGH PIPES.

Absolute pressure of steam in pounds per square inch.	Mean velocities of steam in feet per second.
200	60
100	85
50	120
25	170
$12\frac{1}{2}$	240
6	350
$2\frac{1}{2}$	500

Resistance arising from bends.—The occurrence of sharp bends in a pipe causes serious resistance. When, however, the centre line of the bore follows an arc of large radius, the resistance caused in this way is not important. In all possible cases this radius should be at least equal to three times the bore of the pipe, and in no case should it fall below the diameter of the bore.

Irregularities of surface.—The surface of the bore of a pipe should be as smooth as possible, to diminish the resistance. All roughness, rivet-heads, or irregularities in the fairness of the several lengths where joined together are opposed to the freedom of flow of steam, and they sometimes interfere with drainage of condensed water from the pipes.

Use of cast-iron pipes.—Cast-iron pipes are eligible for use at all pressures up to 90 or 100 pounds above

the atmosphere. Beyond this point they require to be made of excessive thickness and weight. They also cause great loss of heat in radiation, from the increased surface, and in heating the mass of metal after a stoppage. There is a feeling of uncertainty as to their safety, based upon accidents which have happened; but, so far as continuous pressure is concerned, there is no reason why they cannot be made perfectly safe, while they possess the advantage of comparatively low cost. They are, however, more liable than those of other materials to explode from shock, and safety in this respect is of great importance.

Malleable iron or steel pipes.—Wrought-iron and steel are found to be more safe than cast-iron pipes, and they are consequently replacing them largely, at pressures of 100 pounds and upwards, in stationary engine work. They are made from ordinary first quality rolled plates of moderate strength and high ductility, such as are used for boiler work, bent round to a circle, and jointed either by welding or riveting. In many cases a welded joint is covered by a riveted strap, as a precaution against the least possibility of existence of a defect in the weld. The straps are of no less importance than the plates; they should be cut from plates of large size, with the fibre of the plate running circumferentially across the joint, or in the direction of strength against bursting. Plain rolled bars have been thus used, with the fibre running longitudinally, with unfortunate results. The fitting of the strap to the body of the pipe, and the riveting, should be performed with the greatest care. As a rule, the proportions adopted should be such as to avoid the necessity for inside straps, which would be unfavourably disposed for resisting tensile stress. Flush head rivets cannot be closed inside the tubes by hand, but the

heads should be kept as small as possible, and in any case the riveting should be of the highest class. Caulking or fullering inside is quite impossible, and all joints must therefore be closed outside. The ends of the butt straps are especially important, as they are not accessible for caulking after the pipes are flanged. In some cases lap-joints are adopted, but these are not so easily or efficiently fitted with flanges. In good work the butt-joints are welded at each end, even when the chief part of the length is left free for riveting, as it is impracticable to continue the cover-plate over the flange. The riveting may be varied, as in a boiler, according to the strength of joint required.

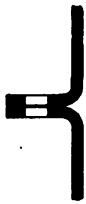

Fig. 90.—Pipe-joint made by turning simple flanges.

Jointing of pipes.—The several lengths of steam-pipes of large or moderate diameters are universally jointed to each other by means of flanges. Cast-iron pipes possess a great advantage in the ease, certainty, and accuracy with which the flanges can be cast on the pipe and fitted together to secure good joints.

Simple turned flanges.—The simplest form of joint for wrought pipes is made by turning a plain flange at each end, as shown in Fig. 90. A little thickness is lost in flanging, and a little more in facing the flanges. Such flanges are therefore found to be very deficient in rigidity, and to prove leaky. The bend of the flange should not be too sharp, or strength will be lost. If, on

the other hand, the curve is too large, the bearing on the surface of the flange inside the bolt-holes is too narrow to allow an efficient joint to be made, and the uniformity of the bore of the pipe also suffers unnecessarily.

Riveted flanges.—A better arrangement than the last is shown in Fig. 91, in which a malleable iron ring is attached by riveting to each end of the pipe. These should be accurately fitted, caulked inside and out, and the end of the pipe faced off with the flange. In this

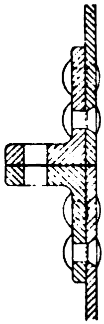

Fig. 91.—Pipe-joint with riveted flanges.

arrangement a good bearing surface is obtained for the jointing material, quite up to the bore of the pipe. The thickness of flange allowed is greater than as in Fig. 90, but is still rather light in some cases, causing leakage.

Welded flanges.—The introduction of electrical welding processes has rendered possible a thoroughly secure and efficient attachment of a substantial flange to a pipe whose longitudinal joint is usually welded by an ordinary process, so that all riveting is avoided. The flanges shown in Fig. 92 are thus attached, and are quite equal to those in cast-iron, with the advantage

of ductile material. Strength, stiffness, soundness, good jointing surface, and freedom from liability to leakage are secured. They may also be arranged with spigot joints, as in Fig. 93, which will prevent the blowing out of joint packing, in case it is desirable to adopt thick soft packing for elasticity. The present disadvantage attached to the use of electrically welded flanges is their high cost as compared with other wrought-iron or steel pipes; but this will doubtless diminish as the practice extends, and other processes are improved to compete with it. T-pieces are very conveniently made of great strength by forging, sawing,

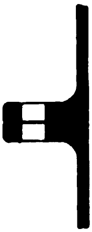

Fig. 92.—Pipe-joint with welded flanges.

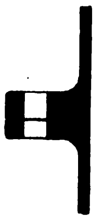

Fig. 93.—Pipe-joint with welded and spigoted flanges.

and electric welding. The practice also gives greater facility than any other for the production of well-rounded angles in the throat of a T-piece, whereby the resistance to the current in the pipe is minimized. The same also applies to sharp elbows, which, however, should only be adopted in absolute necessity.

Copper pipes.—Copper pipes are largely used in marine practice, and to some extent in stationary engine work, especially of small sizes. Solid drawn pipes, up to four inches in diameter, are available, but these are not absolutely uniform in thickness and strength. Large pipes cannot be obtained solid drawn, but are made from plates, bent to shape, and jointed

and provided with flanges by brazing. Numerous disastrous explosions of brazed pipes have occurred, which have led to the addition of bands, brazed round the pipes at close intervals, the joints of which are crossed to clear the joints of the body of the pipe. In some cases a serving of wire is applied to the pipe under a well-defined tension, but this is troublesome, and a slight accident to the wire at any point may affect the whole length between the flanges. There is every reason to anticipate that the use of copper pipes of large diameter will soon cease, in favour of iron or steel with welded flanges, which at present are

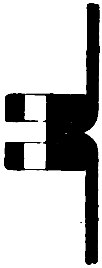

Fig. 94.—Pipe-joint with loose flanges.

almost equal in cost. Small pipes, such as feed-pipes, donkey steam-pipes, or drain-pipes, are best made of copper, whether solid drawn or brazed, which are ductile, easily fitted, and free from liability to serious accident.

Jointing by loose flanges.—Iron and copper pipes, for low or moderate pressures, are sometimes jointed by means of loose flanges, which may be in well-jointed halves, or, if solid, may be placed loosely on the pipe, which is afterwards flanged over, as in Fig. 94. By this means, and with care, a good flange for jointing may be secured. The surfaces are not easily turned in the lathe, and they are liable to damage at any time when asunder. The arrangement gives, however, a

facility for adjustment which is sometimes of special value. Pope's flanged joints for high pressures are strengthened by the addition of an angle-ring to the pipe, against which the loose flange is fitted.

Strength of joint.—The longitudinal strength of a pipe, as regards internal pressure over the main part of the length, is always in excess of the circumferential strength, as in all cylindrical structures. But at each joint the longitudinal strength depends entirely upon the bolts by which the joint is held together. The minimum strength to be provided in these must be sufficient to exceed the fluid pressure over the whole area of the bore of the pipe, and also over the soft packing, or the metallic contact of the joint, as it is impossible to ensure that the internal pressure shall not be applied to the whole. Where the joint is made with soft packing, there must be a sufficient surplus to provide for secure holding of the packing by friction on the two surfaces. The least amount of tensile force which can be allowed for this purpose is six times the amount of fluid pressure upon the exposed inner edge of the packing, which is about twice as much as would usually suffice to allow motion. The disadvantage of thick packing is obvious. The pressure required to ensure tightness in a close joint may be taken as equal to that required in a joint with soft packing a quarter of an inch thick. Bolts for high-pressure steam-piping should be of very high quality, and not loaded above $3\frac{1}{2}$ tons per square inch in area at bottom of thread. Bolts which are just sufficiently powerful to give a combined pressure equal to the sum of the items above given, will suffice to keep a tight joint so long as they bear equally, and the pipe is free from transverse strain, and excessive strain arising from cooling. These conditions are, however, of sufficient importance to call for

an increase in strength, in some cases to double the amount first obtained. For pipes of large diameters, the bolts are sometimes divided over two concentric circles, in which the number of holes are either identical or possess a common factor, the numbers being as 2 : 3, 3 : 4, &c., and no two holes are exactly opposite to each other. The flanges are made of corresponding width. The thickness of every flange must be such as to prevent appreciable bending between the several bolt-holes; in this respect the use of a large number of comparatively small bolts is more advantageous than a smaller number of larger bolts. Rigidity is especially important in flanges made in halves, in which it is best to make each flange in two layers, with the joints separated 90° apart.

Screwed joints.—Small wrought-iron pipes are usually jointed by means of screwed thimbles or sockets, with pairs of flanges at convenient distances for taking apart when required. Union joints are also used with the same object when the operation is frequently repeated. For low pressures running joints may be used, in which the socket is run on one pipe, then back on the second, and secured by lock-nuts. In American practice right- and left-handed screwed sockets are used, ostensibly for facility in taking apart and replacing, but the attendant disadvantages and confusion are more than equivalent to the advantages of the plan. Good wrought-iron pipes of *steam* quality are very convenient for smithing to form, and may be used for any pressure up to 600 pounds. If laid with special care upon a solid bed, they may be used up to 1000 pounds pressure, but they will require constant attention.

Water hammer in pipes.—Pipes of moderate and large diameters are often subjected to severe shocks when at work, which frequently cause a leakage at the joints,

and occasionally a violent explosion. These shocks are usually caused by the existence of pockets or depressions, in which water is collected, which is formed by the condensation of steam during stoppage, or when steam is first admitted to cold pipes. If under such conditions steam is suddenly admitted, the condensed water is violently driven forward, especially if the steam in its course should rise through the water. If there should be any considerable length of free pipe in front of such a current great speed is developed, and at the end of the run the motion of the whole is arrested. The elasticities of the water and the iron are so low that the latter breaks before the motion of the former can be completely arrested. Safety-valves are comparatively useless, owing to the sudden manner in which the pressure is imposed. But with good pipes, provided with an efficient steam-trap at the lowest point, especially if this is at the extremity of the pipe, the danger is not great. It is, however, of great importance that all cold pipes should be gently warmed by the slow admission of steam, to prevent starting of joints. If any suspicion should arise as to neglect in this respect, the use of a watchman's record clock may be adopted.

Expansion and contraction by heat.—In many cases leakage is caused by reason of contraction not properly provided for. The contraction of cast-iron is in approximate accordance with Table XXII.

TABLE XXII.—CONTRACTION OF CAST-IRON IN COOLING.

212°	to 32°	...	...	1 part in 900.
316° (70 lbs. steam)	do.	...	...	1 do. 570.
366° (150 do.)	do.	...	...	1 do. 500.
399° (230 do.)	do.	...	...	1 do. 440.

As an example, each 9 feet length of iron piping, in cooling from the temperature of 70 pounds steam to freezing-point, must contract three-sixteenths of an inch in length, and five such lengths contract nearly an inch. If it were possible to apply a force sufficiently great to entirely prevent contraction, the stress upon the metal of the pipes would reach 14 tons per square inch in sectional area, and that upon the bolts would be greater in proportion to their smaller sectional area. But cast-iron of the highest quality breaks before such a stress is reached. In most cases, a rough balance or mean condition is arrived at, in which a high tensile stress is imposed upon the pipes, or alternately tensile and compressive stress, and a reduced amount of contraction and expansion allowed by reason of the pipes yielding

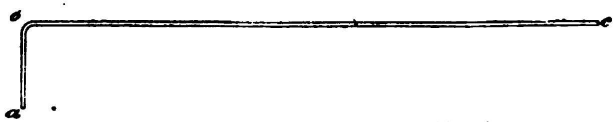

Fig. 95.—Pipe with two arms of unequal length.

at one or more points. With still higher pressures and temperatures the action is more pronounced. Large numbers of joints are persistently leaky from insufficient freedom in this respect.

Disposal of pipes to secure elasticity. — Moderate lengths of steam-piping may be judiciously arranged with two or three right-angle bends of ample radius, which will provide for the requisite expansion by reason of the elasticity of the pipe and joint-packings. Pipes of various kinds are sometimes arranged with bends of U form, which open and close to allow contraction and expansion, but in which excessive bending strains are often imposed. Two arms of piping meeting in a right angle should be as nearly as possible of equal

length. If of very unequal length, as in Fig. 95, and if each end (a and c) is immovably fixed, the bending strain imposed upon the material of the pipe at a will be greater than that at c in the proportion $\left(\frac{b}{a}\frac{c}{b}\right)^2$ on account of the point of junction being bent a greater distance in a shorter length.

Facilities for free expansion.—Stuffing-boxes are the only means whereby the contraction and expansion of long pipes may be efficiently met. This involves the cutting of the pipes, so that the two parts tend to separate with a force due to the pressure of steam in the pipe, and to the area of the turned surface in the stuffing-box. In some cases walls, foundations, &c., are available for the support of the pipes in perfect security, while ample movement is allowed to take place in the stuffing-box. Sometimes the best arrangement is to provide an iron or steel casting in an angle, which gives a good elbow, and two stuffing-boxes, one for each arm. When, however, the stuffing-box for each length or arm is placed near the centre of the length, the movement each way is only half of that which would take place near the box if placed at one end, though the actual sliding in the box is the same in either case. When proper allowance for contraction is made, each angle should be secured, and the intermediate points carried either on brackets provided with dumb-bell rollers, or by rods from above. If no rigid means of attachment are available, a pair of iron rods may be carried parallel to the pipe, under and over, or on each side. These should be secured at each end to strong arms, and to light guides at intermediate points. They should also be protected from considerable heating. Expansion joints which depend upon the flexure of plates also require to be applied under the same conditions, but the

heavy and uncertain strains to which such structures are exposed are sufficient to render them very objectionable for use under high pressure. They are, however, well adapted for pipes working under a vacuum.

Importance of heat expansion.—In connection with piping of all kinds the effects of heat expansion of metals is often insufficiently appreciated until it forces itself into notice. It should, however, be remembered that in railway practice every individual rail is allowed a certain space for expansion at atmospheric temperatures; and every long rod for the operation of switches is provided with a special mechanical joint for expansion.

Application of heat-retaining coverings.—In all cases pipes, stop-valves, and vessels for containing steam, at rest or in motion, should be protected against loss of heat by radiation, and as far as possible by conduction through the supports. Loss of heat in these or in any other ways causes partial condensation of steam, whereby the quantity of true or useful steam is reduced, and the value of the remainder is greatly affected by the presence of the condensed water. Usually the main body of each length of pipe is well covered, but the flanges, bolt-heads, nuts, and a short length of the body are left bare for access, so that any possible leakage may not escape detection. Such surfaces disperse the heat with as much certainty as though they were provided for the purpose. If contraction and expansion were fully provided for, less leakage would occur, and the joints might be covered up. But block or mat coverings, easily removable, may be used for such places with advantage. In every case pipes, whether covered or bare, should be protected from all access of moisture, which cools the pipes by evaporation, and which permeates even the best covering composition, impairing its power of insulation and destroying its substance.

Shifting-pipes.—Pipes of metal are sometimes required to give a little freedom for bending, which may be obtained by the use of a ball and socket or swivelling elbows. The former may be adopted after the manner of ordinary socket-pipes, or by the use of a stuffing-box and gland. In 1888 at Bristol a length of 3 ft. 6 in. piping laid early in this century, with ball joints,

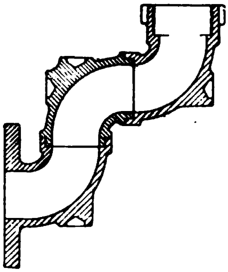

Fig. 96.—Swivelling elbow-pipes.

was easily and successfully lowered over about 150 feet, reaching about 3 feet at the centre, while in continued use under about 15 feet head of water. Swivelling elbows, as in Fig. 96, have been used for 2-inch pipes under 80 pounds steam for many years, the parts being held together by clamps. Similar pipes, held by one central through-bolt, have been also equally successfully adopted for pump pipes up to 18 inches diameter.

CHAPTER XXXV.

INDICATOR DIAGRAMS.

Objects to be secured by means of indicator diagrams.—A correctly-recorded diagram shows the pressure which prevails in the cylinder of an engine when the piston is in any defined position. The mean height of the diagram, measured by a scale of pounds per square inch to the inch upon the diagram, gives the mean pressure upon the piston. The mean pressure in pounds per square inch, multiplied by the number of square inches in the area of piston, and by the number of feet over which it travels per minute, gives the number of foot-pounds of work performed by the piston per minute, which divided by 33,000 gives the indicated horse-power produced in the cylinder.

Conditions necessary to accuracy of results.—The truth of an indicator diagram depends upon the accuracy with which the steam-pressure existing in the cylinder of the engine at any particular moment is reproduced in the cylinder of the indicator, and recorded by the pencil of the instrument. Also upon the accuracy with which the movement of the indicator barrel corresponds with that of the engine piston, so that the record made by the pencil or scribing-point of the indicator is cor-

rectly located as to the length of the diagram. The first condition depends primarily upon the geometrical accuracy of the instrument, but also upon the ample area of the passages communicating between the cylinder and the indicator, upon the moderate length of the same, and upon the freedom from sharp angles and *variations* in the sectional area of such passages. Also upon lightness of moving parts, so that the piston does not move up or down to a greater or less distance than that necessary to record the pressure. The piston of the indicator should fit uniformly all over, sufficiently loosely to avoid appreciable resistance to movement, and sufficiently closely to prevent leakage of steam past the piston of such amount as to influence the pressure in the cylinder. The strength of spring should accurately correspond to the area of piston, and to the divisions upon the scale used in measuring pressures from the diagram.

Connection for rotating the paper drum.—The fulfilment of the second condition above referred to, chiefly depends upon the geometrical correctness of the mechanism employed for the reduction of the motion from the full stroke of the engine piston to the circumferential movement of the indicator barrel. The motion of the indicator barrel is usually derived, directly or indirectly, from the cross-head of the engine. For the sake of lightness, cords are generally used to transmit movement in one direction against the resistance of a spring, which effects the return motion when the pull upon the cord is relaxed. The elasticity of an ordinary cord is much too great for accurate work without special treatment. Hard-spun whip-cord may be used, if heavily weighted before use, and always kept stretched between the occasions of its use. Special cords are made with a view to inextensibility, which is supposed

to be promoted by special lay, or by interlacing of wires. Whatever kind of cord is used, it should be of moderate thickness, as thin cord cannot be relied upon to resist stretching. Fine wire may be used, but is difficult to handle. The spring which effects the return of the barrel should be of strength clearly sufficient to prevent appreciable slackening of the cord, and over-running of the barrel at each end of its stroke, arising from inertia, but any additional strength in the spring, beyond what is thus necessary, tends to cause stretching of the cord, and may cause breaking of the spring. The revolving barrel and its moving attachments should be made as light as practicable, with a view to minimize the inertia. The precise length of the diagram is unimportant when it is in true proportion in all parts of its length. But irregularities in length which arise from any cause are usually confined to one or both end portions, so that the whole is thrown out of scale.

Appliances for reducing motion of engine piston to suit movement of paper drum.—The simplest reducing gears generally applicable are those in which pulleys of different diameters are employed. These are perfect in principle, and may be made practically accurate in work, by a strict observance of the points referred to. Usually, however, the motion is reduced by means of levers. In a beam engine, the parallel motion is most conveniently utilized. In a vertical engine, levers for working the air-pump are generally available, in connection with the indication of one cylinder. But in a horizontal engine there is seldom any detail which can be practically utilized as a lever. Levers must therefore be provided. The arrangement most frequently adopted in horizontal engines comprises a stud supported above the engine, usually by means of a vertical post from the

bed-plate. From this stud (x in Fig. 97) is suspended a lever of length about equal to the stroke of the engine. A pin is attached to some part of the cross-head, and connected by a link to the lower end of the lever, the length of which link should be from two-thirds to three-fourths of the stroke of the engine. The position of the post should be such that the lever will hang vertically—*i. e.* at right angles to the plane of the piston-rod—when the engine is at mid-stroke. The height of the top pin should be such that the mean position of the lower pin is on a level with the pin attached to the cross-head. When the whole is properly adjusted, the pin in the lower end of the lever will assume the same level at each end of the stroke. A second lever or arm is fitted to carry a stud, in such a position as to give the length of stroke required for the indicator barrel. The second lever must be so adjusted that its centre line when at half-stroke will be at right angles to the line of the cord leading to the indicator. If the cord is led away in a horizontal direction, the stud in the second lever will be in the centre plane of the main lever; and conversely, if the stud is placed in such a position, the cord must be led away from the lever in a horizontal direction, guide-pulleys being employed to effect any change in direction to reach the indicator. Fig. 97 shows a gear suitable for an engine of 6 feet stroke, reduced to give a diagram 4½ inches in length, the string being led away at an angle of 1 in 4 with the horizontal. If the stud a were removed to position a_1, and no guide-pulleys used, the centre of the length of the recorded diagram would be incorrectly placed to the extent of 3¼ per cent. upon the total length of diagram, so that one-half of the length of the diagram represents 46¾, and the other 53¼ per cent. of the entire stroke. Sometimes a similarly

defective action arises from tying a string to the edge of a pump-lever or parallel-motion rod. In such cases, the fixed point of the string, which is usually the knot, takes the place of the stud. Pulleys should be applied in such a manner that the pressure upon the pin of each shall be kept as low as possible, so as to minimize the resistance. The length of cord should also be kept as short as possible, to avoid unnecessary stretch.

Telescopic levers are sometimes adopted, but they give a very incorrect action, for the reason that while

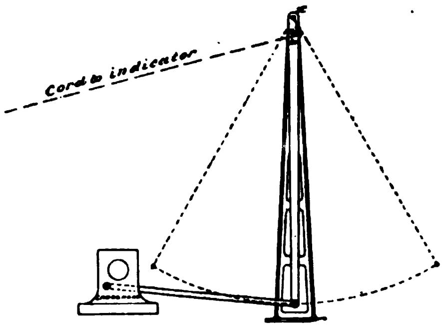

Fig. 97.—Indicating gear, correct.

the length of the short lever remains constant, that of the long lever varies considerably. The greatest length of the lever may be equal to the stroke of the engine, in which case, when the piston of the engine has moved through 25 per cent. of its stroke, the diagram only shows 22·3 per cent. Supposing that steam is actually cut off at this point, the diagram will only record 89 per cent. of the amount of steam actually and usefully consumed in the cylinder, subject to correction for clearance. At an earlier cut-off, or with a shorter lever, the discrepancy is still greater. Such a gear is shown in Fig. 98, from which it will be seen that the ratio

$\frac{b\,c_1}{b\,a_1} = 16$, while the ratio $\frac{b\,c}{b\,a} = 13{\cdot}86$. These, and the corresponding ratios for other positions, should be absolutely equal to each other.

The above examples show the leading geometrical principles which must be embodied in all gears adopted for the present purpose, in order to render the diagrams sufficiently accurate to be trustworthy in use.

Division of diagram and measurement of pressure.—In preparation for estimating the horse-power from an indicator diagram, the atmospheric line should be first ruled in by a fine clear line, if the original one is at all

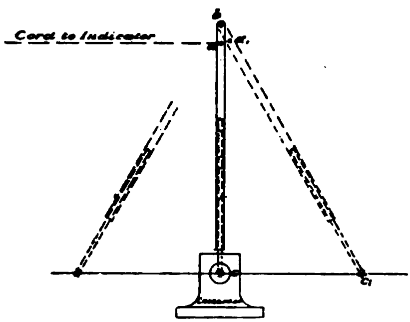

Fig. 98.—Indicating gear, incorrect.

faint. Then draw a line parallel to it, and beneath, at a distance by scale corresponding to absolute vacuum. For accuracy, the distance of this line below the first should represent the barometric pressure prevailing at the time, and at the exact place upon which the diagram was taken, and all pressures should be measured from it. In such a case the atmospheric line would not be required after the line of absolute vacuum was plotted. But simply for convenience, the line of absolute vacuum may be scaled 14·5 pounds below the atmospheric line, which is the mean atmospheric pressure at a level

about 180 feet above the sea. The length of the diagram is next defined by a tangent line at each end, drawn at right angles to the vacuum line and touching the extreme point of the diagram, at whatever part of the diagram this may be. The horizontal distance between these lines is the length of the diagram, and should be equal in the diagrams from the two ends of the cylinder, if produced by the same gear. This distance is to be divided into ten equal parts, by lines continued above and below the diagram. The height in the centre of each strip is then measured by scale, and assumed to give the mean pressure upon the piston during the particular period of the stroke. The mean of the whole ten measurements gives the mean pressure according to the diagram. If, in connection with a simple engine, the several pressures are measured to the absolute vacuum line and the mean obtained, this may be used as the basis upon which the total or absolute power of the engine may be calculated. Part of this power is expended in overcoming the back pressure in the engine; the balance, known as the indicated horse-power of the engine, is available for moving the engine itself, inclusive of pumps and other details, and for performing useful work. In connection with compound or triple expansion engines, the absolute work performed by the engine is obtained by means of the difference between the mean pressure upon the piston and that upon the piston of the succeeding cylinder; this is treated as the actual pressure for horse-power in each case except the last, which is referred to absolute vacuum. The sum of these gives the absolute horse-power of the engine.

Two diagrams to be measured to obtain the actual difference of pressure at any moment on two sides of piston.—The indicated horse-power is sought in almost

every case as the element of chief importance. As already stated, this may be obtained from the absolute or total horse-power, by the deduction of the power lost by back pressure. But it is obtained more rapidly and conveniently by measuring each pressure only from the line of back pressure to the line of steam or positive pressure. The actual pressure upon the piston at any moment is the difference between the back pressure as recorded upon one diagram, and the positive pressure as recorded upon the diagram taken from the other end of the cylinder. This would, however, occupy more time, and obtain only the same total result, as any deficiency at one point is strictly equal to an excess at another point. Sometimes, however, the actual pressure is required in connection with inquiries into the uniformity of turning effort, or strength of parts.

Instrumental measurement of mean pressures.—The measurement of pressures by means of ordinates in the manner described is subject to small error in exceptional cases, as may be seen upon any diagram divided for scaling. Many variations are possible in the shape of the diagram, without affecting the length of the ordinates; and consequently in the power developed, without affecting the power calculated from the diagram by the use of ordinates. These variations are in no case of large amount, and are practically confined to the vicinity of the corners. Several instruments are available, by the use of which such errors are avoided, and time saved in the operation of estimating pressures from diagrams. Such instruments are not difficult to use, but are most successful in the hands of those accustomed generally to the use of mathematical instruments. The one mostly used is Amsler's planimeter, modified for the special purpose.

Pressure at cut-off and terminal pressure.—The

present space does not admit of a full discussion of the conditions affecting the admission line, the steam line, the expansion curve, the exhaust line, the line of back pressure, and the compression line, which are fully dealt with in other works. Two points may, however, be referred to as bearing upon calculations treated in another chapter. The first is the point of cut-off, *c o* in Fig. 99, which should always be regarded as the point at which the supply of steam to the cylinder entirely ceases, and where the curve consequently assumes an opposite character. The pressure in the cylinder may have fallen very much below that shown by the steam

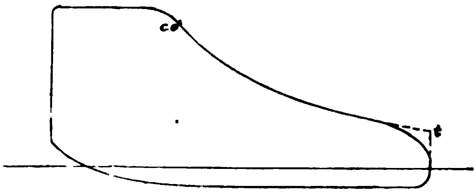

Fig. 99.—Indicator diagram, showing point of cut-off and ideal point of release.

line, but the point of cut-off as described fixes the pressure attained at any subsequent period in the stroke during true expansion. The quantity of steam existing in the cylinder at the time of cut-off is quite definite, and in a series of cylinders working in succession this quantity must remain the same in each successive cylinder, except as may be modified by cylinder condensation, or by leakage of steam, positive or negative. The second point referred to is the point *t* in Fig. 99. No sensible error is incurred in calculations by assuming that the expansion curve is continued to the end of the stroke and the steam completely liberated at that moment, while some operations are much simplified by reason of the assumption.

Combined diagrams of compound engines.—The power to be obtained from a certain amount of steam increases in proportion to the degree of pure expansion to which it is subjected. If it were practically possible to utilize the steam without loss from condensation or otherwise, the amount of work to be obtained from it would be precisely the same, whether the expansion were effected in one cylinder or in several. The initial and terminal pressures being the same in each case, it follows that the last cylinder of a series working compound will be of the same dimensions as the single cylinder of a

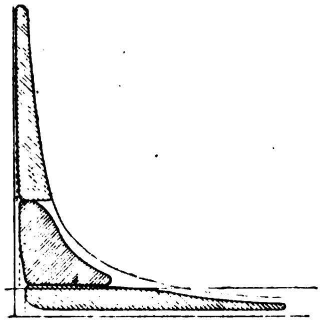

Fig. 100.—Combined diagrams from triple expansion engine.

simple engine, one or more preceding cylinders of smaller size being used to take the first part of the work out of the steam. These conditions have led to the practice of combining the diagrams from successive cylinders, as in Fig. 100, so as to compare the whole with the conditions obtained by means of continuous expansion in one cylinder.

The several diagrams require to be adjusted to the same vertical scale of pounds to the inch, and the length of each diagram in proportion to the area of the piston of the cylinder from which it was taken. In the absence

of clearance in each cylinder, the two, three, or more diagrams may be placed with their admission lines on one vertical line, when the expansion curves will coincide more or less approximately with the unbroken curve of pure expansion. When clearance exists—*i. e.* in all practical cases—each diagram is placed a distance from the vertical line corresponding to the percentage of clearance as compared with the adjusted length of diagram. The vertical positions of the several diagrams will be fixed by scale according to pressure, measured from the same line of absolute vacuum. In connection with triple expansion engines it is often convenient to leave the diagram from the intermediate cylinder unchanged, to increase the length and diminish the height of the low-pressure diagram to correspond, and to increase the height and diminish the length of the high-pressure diagram. Such a diagram in connection with a tandem engine may be usefully plotted. But for any engine in which the several pistons are connected to different cranks there is greater difficulty in the operation, and the result is less satisfactory, owing to the splitting of each cylinderful of steam, and consequent overlapping of diagrams at one point and void spaces at another. The operation becomes especially complicated when two low-pressure cylinders divide the steam, or when two cylinders are used in any other stage. In most cases, the main object with which a combined diagram is constructed can be more conveniently effected by calculation, and comparison of figures.

CHAPTER XXXVI.

ESTIMATION OF, AND TRIALS IN CONNECTION WITH, POWER AND EFFICIENCY OF ENGINES AND BOILERS.

Estimation of power of engines.—From the time of the first use of steam-power, the differences observed in the efficiency of different engines led to a desire for a common basis of comparison. Many of the first applications of steam-power were to the performance of work which had previously been performed by horses. This fact naturally led to the estimation of power by comparison with that of the number of horses displaced. Watt made observations upon the actual power of horses, and found that a good performance of a strong horse was measured by 33,000 foot-pounds per minute. This has remained to the present time as the accepted unit, when the term horse-power is applied to definite work performed. Watt also invented the indicator—in its original form—for recording information as to the working of an engine, upon which it is possible to calculate with accuracy the power developed.

Horse-power, variation in standards.—When Watt considered that the steam-engine had become fully matured, he arranged a system by which the horse-

power could be conveniently estimated from the dimensions of the engine, and the figure thus derived became known as the "nominal horse-power" of the engine. Experience in the use of engines has, however, led to increases in the pressure of steam used, and in the speed of engines, so that an engine of a particular size is now able to perform a very much larger amount of work than is expressed by its nominal rating. The actual amount of horse-power developed in the engine is called the "indicated horse-power," from the fact that it is based upon the average pressure per square inch which the steam exerts against the piston, and which is ascertained by means of the use of the indicator. The difference in mean pressure applied by the steam simultaneously on the two sides of the piston, multiplied by the nett effective area of the piston in square inches, gives the total effective pressure of steam upon the piston, which, multiplied by the number of feet through which the piston moves in a minute, gives the number of foot-pounds of work performed in a minute. When this total is divided by 33,000, the indicated horse-power is obtained. A certain amount of this power is absorbed in driving the engine; the balance is available for performing useful work outside the engine, and is called the useful or effective horse-power. Many makers of engines adopt special standards for nominal horse-power, different from all others. In different districts also there is a great variation. The figures thus arrived at are therefore obviously most unreliable, and fortunately they are rapidly dropping into disuse, except perhaps in connection with agricultural engines, which might with almost equal propriety be sold by the hundredweight.

Duty of engines referred to coal consumed.—The first heavy work done by steam was in the pumping of

mines. Many of these were coal-mines, in which no special difficulty arose as to the supply or cost of the necessary coal. But the coal used in connection with the engines employed in pumping the Cornish mines was subject to the cost of a troublesome sea and land carriage; its consequent high value therefore acted as a powerful incentive to economy. This condition led to the institution of the first standard of definite comparison ever extensively used. This was the number of foot-pounds of power developed by means of the consumption of one bushel of coal. This was a great advance upon preceding methods, but possessed an unsatisfactory feature in the use of a bushel measure. The weight of a bushel varies according to the density of the coal; and a bushel of mixed size is heavier than one of uniform size, whether large or small. Watt estimated the weight to be 84 pounds, but the bushel of Welsh coal subsequently used weighed 94 pounds. In the first half of this century, a duty of 100,000,000 foot-pounds per bushel of coal was exceeded. This is equal to 1·86 pounds of coal per effective horse-power per hour, the duty being estimated upon the supposed amount of water actually pumped, and not upon the indicated horse-power. The latter must be greater than the former, and consequently the consumption per indicated horse-power would be appreciably smaller. This standard would appear to be more suited for application to pumping than to any other work, and it does not appear to have been much adopted except in Cornwall. In the abstract, however, there is no reason why the power of all engines should not be estimated in the same way, provided that the coal be stated in weight instead of measure. Both this and the indicated horse-power are equally based upon the foot-pound as an original unit. In the great majority

of cases, however, comparison is made upon the weight of fuel consumed per hour per indicated horse-power. In all cases, notes as to coal consumption should include information as to the quality of the coal. The most valuable element in this connection is the calorific power. But this is not available to the extent which is justified by its importance, and in its absence an estimate of the value of fuel may be based upon its chemical composition as obtained by analysis. In the same district, the calorific value of steam coals may be fairly assumed to vary in approximate accordance with the selling price. For this reason some engineers adopt the very convenient standard of cost of coal per 1000 indicated horse-power hours, but no very simple means exist whereby the whole of the required information may be conveyed.

Engine and boiler trials separately conducted.—Though comparisons only based upon the coal consumption and the indicated horse-power are of considerable interest and value, they utterly fail to give any sign of excellence or defect in detail. In 1839, Mr. Parkes said that this system "mystifies the subject, tends to obscure our knowledge of the actual degree of excellence attained by the steam-engine, and retards the extension of the most effective system of applying steam; the generation and the application of steam being distinct problems, require to be treated separately." Many engineers have given great attention to the subject, and the practice of independently testing boilers and engines is now well established. These trials are directed to give information on five chief questions—

I.—The total amount of heat which is due to the quantity of coal consumed, and to its full calorific power.

II.—The amount of heat generated in the furnaces.

III.—The amount of heat transferred to the water in the boiler, inclusive of economizer or feed heater.

IV.—The amount of heat transformed into useful work in the engine.

V.—The amount of heat rejected by the engine and lost.

Quantitative information upon many other points is collected and recorded, some entering into the calculations, and some for purposes of comparison with the results of other tests, or to explain any abnormal conditions which may be observed.

Consumption of coal.—The weight of coal is ascertained by careful weighing throughout the progress of the trial. The quantity of ash produced in the trial is also weighed; and the quantity of unconsumed carbon lost in the ash is ascertained by chemical analysis. The amount of heat which the coal is capable of producing if burnt to the last particle, is calculated upon the results obtained by actual combustion of a small sample in a calorimeter, as described in another chapter. This should be verified by comparison with the results of a chemical analysis, and the experiment repeated, if not fairly supported by the analysis. The possible amount of heat which can be obtained is subject to reduction on account of carbon lost in the ash, and of the production of carbonic oxide, when this is detected in the analysis of gases. The fire should be left in the same condition at the end of the trial as at the beginning.

Estimation of air supplied.—Information as to the quantity of air supplied to the furnaces is essential to the satisfactory dissection of the results of a trial. Owing, however, to the practical difficulties involved in the adoption of any available process of direct measure-

ment, the quantity is deduced from the results of analysis of furnace gases.

Estimation of heat in gases.—Assuming that combustion is quite complete, the whole of the carbon burnt is discharged as carbonic acid, the weight of which is obtained from the weight of coal actually burnt. The weights of carbonic oxide, oxygen, and nitrogen are calculated upon their respective volumes as compared with that of carbonic acid. Alternatively, the same items may be divided into carbonic oxide, surplus air, and nitrogen. In the latter case a quantity of nitrogen = quantity of oxygen × 3·805 should be separated from the total nitrogen and added to oxygen to form air, the remaining nitrogen being considered alone. Water, vapour, or steam also occurs in the gases, being produced in the combustion of the hydrogen in the fuel, and also supplied in the water existing in the coal, or as vapour in the air supplied. The weights of the several constituents per pound of coal burnt are calculated, and multiplied by the several specific heats, so that the total capacity for heat per degree of temperature of the mixed gases is obtained. From this and the difference in temperature between the air supplied and the chimney gases at any point, the amount of heat still remaining in them may be directly obtained, and that already surrendered may be obtained by subtraction; and finally, the amount of heat which has been produced in the furnaces, and ultimately carried to waste up the chimney. The item of heat lost in the chimney gases may, in the report, be stated in one amount or divided into several.

Temperature observations upon the gases.—The temperatures of the gases are observed by means of mercurial thermometers, filled with nitrogen above the mercury. On the expansion of the mercury by heat,

the nitrogen becomes compressed, whereby the boiling of the mercury is prevented, up to temperatures of 800° to 900° F. The most important temperature observations upon the gases are taken between the boiler and the economizer, and between the economizer and the chimney, so that the amount of heat imparted to the water in the economizer may be separated from that imparted to the water in the boiler. In the analysis of results these should be completely dealt with, both in combination and separately.

Measurement of feed-water and its contained heat.—The quantity of the feed-water is usually measured in a large vessel, the contents of which, under working conditions, have been weighed, and a hook-gauge or other device adopted for showing accurately the level to which the water should be filled. In many cases, two such vessels, alternately used, are necessary. The temperature of the feed-water is taken before entering the economizer, and also on its way from the economizer to the boiler. The quantity of heat absorbed by the water in the economizer is then compared with the quantity simultaneously surrendered by the gases on their way to the chimney, and should be found to nearly balance.

The quantity of water fed into the boiler is one of the most important elements to be decided in the trial, as it affects a very large proportion of the quantities dealt with; and most unquestionably the most reliable information is to be obtained by subjecting the whole of the feed-water supplied to actual weighing test, though it is usually only measured.

Proportion of unevaporated water existing in steam.—The precise amount or absence of unevaporated water in the steam is of great importance. A sample for testing as to this condition should be collected in such

a manner that it is neither more nor less dry than the bulk. The sample is usually tested by condensation, as described in another chapter. The presence of any large amount of unevaporated water in the steam produced by large and well-proportioned boilers is now very exceptional. But any which may occur affects the apparent relative efficiency of the boiler and engine, and also the absolute efficiency of the whole; hence the importance of its accurate estimation. In its absence, credit may be inadvertently claimed on behalf of the boiler, for the evaporation of water which is only primed off, which is of no use to the engine, but which, by its action in promoting or increasing cylinder condensation, is prejudicial to the performance of the engine. The measurement of the heat discharged by the air-pump administers a check upon this item, but does not furnish the means for its direct measurement.

Balance of weights of water and steam, also of heat.— When precautions are adopted to prevent leakage, when blowing-off is either avoided or allowed for, and when the quantity of water in the boiler is not allowed to fluctuate, the total weight of steam supplied by the boiler, inclusive of any suspended water which it may contain, is equal to that of the water supplied to the boiler. The amount of heat absorbed by the water in the boiler and economizer, together and separately, may be calculated from its weight, and the quantity of heat contained by one pound before and after exposure in each case. The quantity of heat contained in steam and water respectively, at the temperatures corresponding to different pressures, are shown in Tables IX., X., and XI., from which may be deduced the amount of heat contained in any mixture of wet steam, of which the proportions are known. The ratio of the amount of heat absorbed by the boiler to the total amount due to

perfect combustion of the whole of the fuel consumed is the co-efficient of efficiency of the boiler. The ratio of the heat absorbed by the economizer to the amount of heat remaining in the gases on reaching the economizer is the co-efficient of the economizer alone. A third co-efficient refers to the boilers and economizer as one instrument.

The difference between the total amount of heat expended and that utilized is accounted for by the following losses—

I.—Heat carried away in the gases arising in the process of combustion of the amount of coal consumed. This is inclusive of the nitrogen corresponding to the air actually consumed, and the water vapour contained in such air.

II.—Heat lost in the surplus air passed through the furnace, and in the vapour contained in such air.

III.—Undeveloped heat on account of partial combustion of fuel, as shown in the analysis of gases.

IV.—Heat lost by reason of the temperature of ashes when drawn from the furnace, and undeveloped heat lost by reason of the presence of unburnt coke contained in ashes.

V.—Heat unaccounted for, chiefly lost by radiation of surfaces of boiler and brickwork, and conduction from the same into the ground.

The amount of last item may be approximately estimated from the amount of coal required to just maintain steam-pressure when shut down. The amount thus determined is rather less than that arising during full work, but no better method is available. When this amount is added to items I. to IV., and to the amount of heat utilized, the total will be usually found

to be somewhat less than the amount of heat expended. But in some cases it appears to exceed the total amount of heat due to the coal consumed. This arises from some error, of which the most probable is that unevaporated or priming water, carried off in the steam from the boiler, has escaped cognizance in the test, which should be repeated.

Absolute zero of temperature and of pressure.—The absolute amount of heat contained in fuel—as in all other substances—depends upon the temperature as calculated from absolute zero of temperature, which is 461° below the zero of Fahrenheit's scale. All calculations as to the expansion of perfect gases by heat must be based upon absolute zero, which is also essential to the satisfactory treatment of many other questions. But this point has no present connection with the economical application to heat, in which a definite quantity of heat is available to be utilized at a temperature above that of the atmosphere and surrounding objects at the time, which quantity cannot exceed that produced in combustion. Many calculations, mental and otherwise, would however be facilitated if all pressures were referred to an absolute vacuum, instead of to the pressure of the atmosphere, which varies largely at different times and in different places.

Heat converted into work.—The amount of heat converted into work obviously depends upon the amount of work done, or the indicated power, which must be obtained with great accuracy. Indicators are employed to take simultaneous diagrams from the several ends of the cylinders. These are repeated at close intervals, and an average taken from the whole. Every precaution should be taken to keep the engine at a uniform speed at and about the time of taking diagrams. At any moment when the speed is increasing, the indicated

horse-power is in excess of the work to be done, and conversely, when the speed is diminishing the work is in excess. As a moderate example of this, an engine with a fly-wheel whose rim weighs 60 tons and runs at 5000 feet may be considered. If the speed of such a wheel is increased or diminished by one per cent. during the space of one half-minute, the power recorded by the indicator is nearly 18 horse-power more or less than sufficient to drive the load at the time. But much greater differences than this arise frequently. Such difference is absolute under all conditions of loading, but assumes more importance when indications are taken to give the friction due to unloaded engines. At such times the absolute variation bears a greater proportion to the whole, and the engine is much more difficult to control at uniform speed. In an extreme case steam may be entirely or partially shut off, so that the engine may come to rest. Indications taken at such times are obviously worthless, but only vary in degree from those taken under ordinary variations of speed. The number of foot-pounds of work, as shown by the diagrams, is reduced into indicated horse-power on the one hand, and on the other into equivalent units of heat, which are compared with the units of heat supplied to the engine.

Heat discharged in water and air.—The quantity and temperature of any water drained from steam-jackets are ascertained, and also any heat lost otherwise—as by air discharged from jackets—is either measured or estimated, and recorded. In the measurement of small quantities of water, Mr. Donkin uses a tip-can which contains 100 pounds of water at 60° F.—though any other convenient quantity and temperature may be similarly adopted. When used for water of any temperature other than that intended, a suitable correction must be made. The chief points of convenience in the use of such a can

are the accuracy with which it may be filled, and the facility with which it may be emptied. Here, as in other operations, still greater accuracy is secured by weighing the water instead of measuring it.

Heat rejected in water discharged from the condenser.—In all cases a large proportion of the heat imparted to the steam in the boiler and supplied to the engine passes away in the exhaust steam to the condenser, or into the atmosphere. The amount of this heat varies in different cases from 80 to more than 90 per cent. of the total heat supplied, which causes its exact measurement to be an object of great importance. On account of its simplicity, the case of a surface condensing engine may be first considered. Here the steam loses an important proportion of its heat by reason of the work which it performs, equivalent to 42·75 thermal units per minute per horse-power of work done. The chief part of the remaining heat is imparted to the condensing water, which is passed through the tubes of the condenser, the steam to be condensed being directed into contact with the outside of the tubes. The amount of heat thus lost by the steam is precisely equal to that acquired by the condensing water, and may be measured by the quantity of water and its increase in temperature. The quantity of water may be approximately ascertained from the capacity of the circulating pump; but it is usually measured by passing it through a box, and discharging it over a gauge notch or through a gauging orifice. Diaphragms are fixed in the box in such a manner as to still the surface, and permit of accurate readings of the surface level, from which the quantity may be calculated or found by table. The quantity of condensed water discharged by the air-pump may be found in the same way or weighed, and compared with the quantity of feed-water. The heat remaining in the air-pump

discharge is not lost if the whole is fed into the boiler without reduction of temperature. Otherwise, an amount of heat is lost, in proportion to the quantity of water and the difference in temperature between the feed-water and the air-pump discharge. This is added to the heat lost in the condensing water, and the total stated in units per minute, which is divided by the number of indicated horse-power. This gives the number of units of heat per minute per indicated horse-power rejected by the engine, which is an important co-efficient or standard, by which comparisons may be made between different engines, or between the same engine at different times or under different conditions. In tested engines this has been found to vary between 200 and 500, and many untested engines would far exceed the latter figure. Obviously, the most economical engine is the one in which the amount of rejected heat is least. If the engine possesses steam-jackets, it must be debited with the steam supplied to them, and the heat discharged from the jacket-drains measured and added to that discharged from the condenser.

Heat rejected from a jet condenser.—In the case of a jet condensing engine, the condensed steam and the condensing water are mixed together, but the manner of treating the whole will be obvious from the description of the method of procedure with a surface condensing engine.

Weight of live steam at successive periods obtained from volume and pressure.—The quantity of live steam in any cylinder of the engine at any time may be measured by the nett sectional area of cylinder and the length of stroke completed at the moment, with a suitable addition for the amount of cylinder and port clearance. In the sectional area of cylinder, allowance must be made on account of the area of piston-rod with

suitable regard to whether the rod is continued on both sides of the piston. The pressure of steam at the same moment is obtained from the indicator diagram, and the weight of steam obtained from the volume and the weight per foot, as given in the tables. This is generally calculated at two points for each end of each cylinder, one of which is at, or immediately after, cut-off, and the other is at, or immediately before, release. By this means the weight of absolutely dry steam is obtained. But owing to causes explained in another chapter, the quantity of dry steam thus derived is always found to fall considerably below the weight of water supplied to the boiler, and known to exist in the engine at the moment. The difference may be taken as unevaporated water, diffused in the steam, or attached to the surfaces of the cylinder. The amounts of heat in the steam and in the unevaporated water may be found by the use of the tables. The total heat of the mixture of steam and water may thus be determined, and the successive conditions as to disappearance of heat may be compared.

Heat lost from surfaces.—The amount of heat transformed into work, and that rejected by the engine through the condenser, when added together will not quite equal the amount of heat supplied to the engine. The chief cause of difference is the radiation of heat from the surfaces of the cylinders and pipes. As in the case of the boiler radiation losses, this may be approximately estimated upon the amount of steam condensed, in maintaining the whole at working temperatures when the engine is standing.

Causes of variation in amounts of rejected heat.—The heat furnished by one pound of steam per hour will raise the discharged heat about 20 units per minute. If therefore, in any particular case, the co-efficient is found to rise either gradually or suddenly by this

amount, it may be reasonably assumed that the consumption of steam has increased by one pound per horse-power-hour. This may probably arise from leakage of steam, or from cylinder condensation. A loss, generally smaller in amount, may arise from the supply of the steam in a wet condition. In this way, the presence of one pound of unevaporated water per horse-power-hour, in steam of 45 pounds absolute pressure, would raise the rejected heat by 3·6 units, and in steam of 215 pounds absolute pressure would raise the rejected heat 5·5 units. The rejected heat is correspondingly increased by heat obtained from steam, which is condensed at once upon its entry into the cylinder, and re-evaporated during the period of exhaust.

Difficulties in connection with estimation of large quantities of rejected heat.—The above system of testing engines of moderate power has been practised with great success. But the difficulty involved in fitting the gauging apparatus of dimensions to suit engines of 1000 horse-power or more has hitherto prevented its application on such a scale. Owing to the trouble involved in, or in some cases the impossibility of, measuring accurately the heat lost by radiation and in sundry ways, the two are taken together as equal to the whole amount of heat supplied to the engine, less by the amount converted into mechanical work, or 42·75 thermal units per indicated horse-power per minute.

Provision of appliances upon new engines for convenient measurement of rejected heat.—Though the provision of an ordinary testing-box suitable for use with large powers, and the requisite alterations to pipes in connection with the testing of an existing engine, is a costly undertaking, there appears to be no good reason why gauging appliances should not be provided in the erection of new engines. In many

instances, however, two cooling-ponds are provided, when one may be arranged for use as a measuring tank while water is drawn from the other one. An accurate float-gauge, connected with the pond by a small pipe or a small hole, and provided with a vernier for close reading, will give the level in the pond to within one-hundredth part of an inch. This, taken in connection with the area of the pond, will give the quantity of water discharged into it, correction being made on account of temperature. When the choice lies between two ponds otherwise equally convenient, the smaller one should be selected, on account of the possible greater accuracy in measurement and the less loss by evaporation. The amount of loss is dealt with in the chapter on cooling-ponds. But an opportunity should be taken to measure the amount of this, and of any possible leakage, for a few hours, and apply the results to the observations made during the trial. Such determination should be made and repeated when the surface water in the measuring pond is, as nearly as possible, at the same temperature as in the trial, and the atmospheric conditions are also similar. The superficial area of the pond should be estimated with accuracy, and allowance made on account of the sloping sides increasing or diminishing the area.

The adoption of minor trials frequently repeated.—Complete and systematic trials as just described are chiefly made by Inspection Associations, or by makers of engines. But though the staff available on ordinary works is quite insufficient for such a task, there are few cases in which it is not possible to secure great advantages at a most moderate expenditure of time, and still less of pecuniary cost. The frequent use of the indicator and attention to the results are so universally recognized in good practice, that it is unnecessary to enlarge upon

them. In connection with the engine, the next important provision to make is for the gauging of the heat lost in the air-pump discharge, chiefly for the reasons in this connection that no other means are available whereby concealed leakages or defects declare themselves, and because this test gives the means to compare the working of the engine independently of the boiler, which again is often convenient on account of steam being taken from the boilers for purposes other than driving the engines. If an air-pump with open top is used, and there is a deficiency in fall, a box as already described may be used. But if no difficulty arises as to the available amount of fall, a vessel to discharge horizontally through a gauged orifice, and provided with a vernier arrangement for accurately observing the vertical distance from the centre of the orifice to the surface of the water, is more convenient. Hitherto these appliances have been temporarily applied, and carried from place to place as required. But an apparatus of this kind permanently attached to an engine would well repay its cost. In each case an appropriate size of orifice would be required, and a table prepared to suit it, with columns for temperature, and lines for the several heads of discharge corrected for the type of vessel adopted. A large water-meter of open type such as Parkinson's, or a good piston water-meter such as Kennedy's, would also measure the water with fair accuracy. But the *absolute* quantity can only be obtained by weight, for which purpose automatic or manually operated weighing-machines of the turn-over class, as used for weighing grain, would be found to be well adapted. Systematic temperature observations present very little difficulty of any kind, and are of great service. These should be chiefly directed to the flue gases and the feed-water, in each case being taken

before entering, and after leaving, the economizer. The quantity of feed-water may be obtained from a permanent water-meter, but a weighing-machine would be better in this case also when applicable. The size of the appliances required for measuring the quantity of feed-water is much smaller than those for the air-pump discharge, on account of the less quantity of water to be dealt with. The amount of heat absorbed by the water in the economizer is obtained from the actual quantity observed to pass during a definite time, and the amount by which its temperature is increased. Averages taken from a week's work are useful, but are not precisely accurate, owing to the usual and unavoidable irregularities of feeding. The amount of heat imparted by the gases is not easily and directly measured, but it is equal to that simultaneously received by the feed-water. The temperatures of the gases are chiefly of value in showing whether the heating surfaces of the boiler and of the economizer are in efficient operation.

Periodical analysis of gases.—The advantages presented by the practice of analyzing the furnace gases are not so easily and simply secured as those obtained by the means just referred to. But they are no less important, and do not present any serious difficulty. It cannot be supposed that all persons in charge of steam machinery will be able to make such a complete and absolutely accurate analysis as a chemist in full practice. But the manipulative skill required for making a very useful volumetric analysis is not greater than that required in the production of a fairly good photographic negative, in which so many amateurs excel. This practice is certain to be often useful as a check upon the system of stoking and upon the personal qualities of the stokers, and is absolutely essential to a satisfactory treatment of such questions as mechanical

v. hand-firing, or the economical qualities of appliances for the prevention of smoke. The time necessarily occupied is most inconsiderable, and the outlay for apparatus comparatively trifling. Tests should be frequently repeated, independently of apparent special necessity, by which means efficient working will be promoted.

Current practice.—Messrs. Farey and Donkin introduced the system of detailed trials into English practice, and the first reported case appears to be one made at Hele in Devonshire in 1871. Recently, however, many such trials have been reported in the technical papers, giving many details. But in comparatively few cases is the amount of rejected heat reported to have been accurately and directly measured, though it is by far the largest item of heat which enters into the consideration of any case.

References.—Reports of trials in which the performances of the engines are given in great detail, and those of the boiler in less detail, will be found in two papers by Mr. Mair-Rumley in the *Minutes of Proceedings of the Institution of Civil Engineers,* vol. lxx., p. 313, and vol. lxxix., p. 323; also in papers by Mr. Willans, published in *Minutes of Proceedings of the Institution of Civil Engineers,* vol. xciii., p. 128, vol. xcvi., p. 230, and vol. cxiv., p. 2; also in a paper by Prof. Reynolds in the *Minutes of Proceedings of the Institution of Civil Engineers,* vol. xcix., p. 152. These, and the discussion upon the three last-mentioned papers, will be found to be of great interest.

Precautions in advance of trial.—Before the time of trial, stop- and safety-valves which may require to be closed should be ground tight; or if this should be impossible, means must be provided for measuring any loss which may arise. Safety-valves should be adjusted

correctly, so that by no probable means shall steam be lost by premature blowing off. All stuffing-boxes should be examined before the test, and re-packed if not found to be in good order. Blow-off valves and other connections which are not required to be available for use during the time of trial should be closed by blank flanges. If any such are required for use during the trial the quantities of steam or water withdrawn must be measured; also the temperatures and condition of the same. As a rule, one boiler is separated during the trial for the supply of steam for purposes not covered by the trial.

Descriptive particulars of plant required.—All significant dimensions are required to be accurately measured and descriptions noted. These include the dimensions of grates and heating surfaces, description of boilers, diameter and stroke of each piston, the dimensions of ports and pipes—including particulars of angles traversed—volumes of clearance, diameters of piston-rods, bore and stroke of air-pump, particulars of feed-pump, and of circulating pump, if one exists. These should be taken from the actual parts themselves as far as possible, and not from any records. The condition of each piston, of the surface of the bore of each cylinder, of the slide-valves and facings, to be examined and tested under steam-pressure in different positions, and found good or attended to. The condenser and connections should be tested for inward leakage of air; and in the case of a surface condenser both tube-plates should be examined, or the entire condenser tested by water under a head of a few feet. All examinations as to condition should be made shortly before the time of the trial, to ensure the existence of good working conditions at that time, and great care exercised in replacing the several parts. Naturally, the required dimensions and particulars

will be ascertained and recorded at the same time of opening the parts. The indicating gear should be examined and the proper length of diagram in each case ascertained, which should be checked with the actual diagrams recorded. If these are found to be too long or too short, the reason should be ascertained and rectified. The indicator cocks and connections should be examined as to sufficiency of area of thoroughfare and freedom from sharp bends. Any detail whatever which may not be in reasonably good condition should receive due attention. One reason for this is that the results obtained in testing plant which is out of order possess absolutely no value, except to compare with the results of trials made upon the same plant when in good order. Another reason is the probability that one point may not check another in a satisfactory manner, unless the whole is in good order, a condition which detracts very much from the confidence accorded to the results by the conductor, and much more so by other persons.

Examination of boiler flues and pipes.—The boilers should be at least moderately clean, and a responsible man should be sent through the flues on each side of each boiler, and around the economizer to the main damper to ensure that everything is in good order, and that the existence of no detail is overlooked. For the same reason he should follow the course of the main steam-pipe and of all branches, the boiler feed-pipes, blow-off and scum-pipes, and all cylinder and pipe-drains; and he should generally become personally acquainted with the existence and condition of every detail. If any branch is found to be connected to either of the above pipes, it should be disconnected or followed to its termination, and the amount of any discharge or suction measured. All parts which cannot be con-

veniently examined while under pressure should receive special scrutiny as to signs of leakage which may exist.

Verification of instruments.—Before and after the trial, all pressure-gauges, indicators, and indicator springs should be compared at working temperatures with standard pressure-gauges and verified, the latter also with the actual scales used for measuring the diagrams. All thermometers should be occasionally compared at several temperatures with standard thermometers, preferably air thermometers, and especially is this necessary with those thermometers in which compressed nitrogen is used.

Records.—The records of trials should show precisely the plant tested, where situated, when constructed, and the date and length of time covered by the trial. Also the date and results of previous or subsequent examination of details. The meteorological conditions should be noted, the particulars of the chimney, the work done, and as much additional information as may arise from time to time, and which may by any possibility become useful.

Possible causes of reduction in efficiency.—When the amount of coal consumed in any particular work is observed to rise without apparent reason, it is usually due to one of the following causes:—

I.—Depreciation of quantity of coal.

II.—Reduction in evaporative power of boiler.

III.—Development of a defect whereby steam is lost by leakage, or is imperfectly distributed in the engine.

IV.—Leakage or defect in the condenser or air-pump.

V.—Increase in the indicated horse-power, due to increased friction, probably arising from defective lubrication, or from a structural settlement, of such a nature as to impose abnormal strains upon moving parts.

Useful power measured by brake.—Tests of small engines usually include measurement of the useful power produced, which is very conveniently effected by means of a brake, which, by means of solid or fluid friction, opposes an adjustable resistance to the motion of the engine, and causes the conversion of the whole of the mechanical work into heat. The brake is provided with a lever, to one point of which is applied—at right angles to the line joining the point of application to the centre of the shaft—a stress of amount sufficient to prevent motion, which tends to take place by reason of the revolution of the shaft. The amount of this stress, in pounds, multiplied by the space in feet through which the point would move in one minute if allowed to do so, gives the foot-pounds of work performed in one minute by the engine, which divided by 33,000 gives the effective horse-power of the engine. The indicated horse-power is simultaneously obtained, and exceeds the brake horse-power by an amount which is equal to the power required to drive the engine under the conditions. This is rather greater than would be required to drive the engine when quite unloaded. In some minor cases the heat developed in the brake is allowed to become dissipated without the adoption of special measures to promote it. But in ordinary cases this heat is carried off by water, oil, or other fluid, either freely poured upon the surfaces, or circulated through channels specially provided for the purpose. The number of thermal units of heat developed per minute may be obtained by multiplying the effective or brake horse-power by 42·75. Its direct measurement is, however, a very difficult matter, by reason of the number of parts over which it is distributed, and the constant loss which takes place by diffusion in the air or over surrounding parts.

Difficulty in application of brake test to large powers.—The brake test is practically inapplicable to large engines fully loaded, on account of the cost and inconvenience involved in the provision of the requisite appliances. But occasions may arise on which it may be desirable to apply such a test to large engines when lightly loaded. A brake test, in which 178 horse-power was obtained, is reported in *Engineering*, vol. xxv., pp. 4, 21.

Apportionment of powers in separate items.—Efficiency trials are generally made upon engines when working at ordinary full power. In all cases it is, however, desirable to ascertain the proportions of power absorbed by the engine, gearing, and all the main shafting, together or separately. Also the power consumed by the machinery employed in each department. At the times when such trials in detail are made, the belts in all possible cases, other than those in the particular department under test, should be removed from the pulleys. Results thus obtained will often prove useful for future comparison, especially in cases of observed increase in the total indicated horse-power. Such results are more reliable when obtained in a connected series of observations, and special care is required to maintain uniform speed. But every opportunity should be embraced for repeating them, at such times as separate departments may be in full operation without the rest. In trials upon pumping engines, the actual work done admits of measurement, approximately at least. In air-compressing and refrigerating engines, a large amount of energy is directly transformed into heat, which enters into the balance of account.

Uniformity of report for purposes of comparison.—Every series of trials is followed by the presentation of a detailed report, which deals only with facts absolutely

obtained at the time, but in which reference may with advantage be made to other reports which may have been made upon the same plant, or in which results may have been obtained which have a special bearing upon the case in question.

Epitome of report.—But in addition to the detailed account of constructive data, meteorological conditions, results of observations, and deductions from the whole, an epitome should be presented in a form as nearly uniform in all cases as it is possible to be, and readily understood by any person without the possession of any special technical knowledge. This should be such that useful comparison with other plant, or with the same plant at other times, may be at once drawn, and hence it is desirable that the form should cover most cases without change. Obviously, such uniformity cannot extend to the bulk of the report, though even here simplicity and uniformity should be followed as far as possible. Such an epitome will best fulfil its purpose when the chief portions assume a tabular form, and when the questions of apportionment of heat in boiler and expenditure of heat in engine are covered by balance-sheets arranged similarly to such as deal with money values. Forms are given in Tables XXIII., XXIV., and XXV., which will suffice to show the essential points in such a system. Figures indirectly obtained are distinguished by being placed in brackets, all others to be understood to be based upon direct observation.

Example to illustrate the importance of independent trials of boilers and engines.—As bearing upon the great importance of a separation between the performances of the boiler and the engine, an instance may be assumed in which two sets of boilers and two engines are employed in two separate installations, in

each of which the combined result is the production of 1 horse-power by the consumption of 2 pounds of coal. In the one case the boilers may actually evaporate 10 pounds of water per pound of coal, while the engine consumes 20 pounds of steam per indicated horse-power. In the other case the boiler may, on account of the absence of an economizer, or for other reasons, only evaporate 7 pounds of water per pound of coal, while the engine only requires 14 pounds of steam per indicated horse-power. A striking comparison would be made by supplying the steam from each set to the alternate engine, when one of the new combinations would result in the consumption of 1·40 pounds of coal per indicated horse-power, and the other in 2·86 pounds, or more than double the first. By improving the defective boilers and the defective engines to equal the others, a consumption of 1·40 pounds all round would be secured. In this example round figures are taken, but not more extreme than are often met with in practice.

TABLE XXIII.—ADDRESS. DATE. DURATION OF TRIAL.

Two Lancashire boilers, 30 ft. × 7 ft. 6 in. diam. Furnace-tubes, 3 ft. diam. Mechanical stokers used. Grate surface, 36 sq. ft. per boiler. Heating surface, 1,040 sq. ft. per boiler. Green's economizer, 192 pipes, 1,920 sq. ft. total heating surface. Chimney, 180 ft. high; area at outlet, 2,700 sq. in.

	per cent.
Analysis of dry coal: carbon 80·56, hydrogen 5·02	85·58
Sulphur 1·30, nitrogen 1·36, oxygen 9·06 ...	11·72
Ash	2·70
	pounds.
Air supplied per pound of dry coal burnt ...	20·50
Dry coal burnt per square foot of grate	15·23
Pure steam produced, per pound of dry coal burnt	9·30
Equivalent evaporation from and at 212° ...	10·80
Water evaporated in boiler per square foot of heating surface	4·81

Draft at foot of chimney. ·80 inch column of water.

BALANCE-SHEET OF HEAT APPORTIONMENT PER POUND OF COAL.

British thermal units.		B. T. U.	per cent.
Calorific value of one lb. of dry coal = 13,843	Heat imparted to water in boiler, 9,473		
	Heat imparted to water in economizer, 1565	11,038	79·65
	Heat lost in the gases produced in combustion	976	7·04
	do. surplus air	651	4·81
	Heat undeveloped by reason of imperfect combustion ...	75	·54
	Heat lost in ashes	833	6·01
	Radiation and heat unaccounted for	(270)	(1·95
Total 13,843	Total	13,843	100·00

POUNDS OF WATER ACCORDING TO MEASUREMENT OF BOILER-FEED.

	Per pound of coal.		Actual per indicated horse-power per hour.	Per cent.
	Actual.	From and at 212° F.		
Evaporated	9·27	12·35	13·39	97·0
Primed	·28	·38	·39	3·0
Total	9·55	12·73	13·78	100·0

TABLE XXIV.

Triple expansion engines, three cranks. Steam cut-off by gear of Corliss type. Power used for cotton spinning. Ordinary limits of work, 15 per cent. below and 8 per cent. above average. Stroke, 5 feet in all cylinders. Piston speed, 700 feet per minute. Weight of dry steam per indicated horse-power per hour, 13·39 pounds. Heat rejected from condenser, 214·4 units per indicated horse-power per minute.

PARTICULARS OF CYLINDERS.

	Diameters.	Areas.	Ratios of total volume, including clearance.	
	inches.	square inches.		
High-pressure	$16\frac{1}{32}$	194·75	1 : 2·64	1 : 6·89
Intermediate	$26\frac{1}{16}$	526·39		
Low-pressure	$42\frac{1}{16}$	1,882·47	1 : 2·60	

	Temperature.	Absolute pressures. Actual.	Absolute pressures. Mean effective.	Cut-off.	Indicated horse-power.
	F.				
Boiler	368·1°	169·9			
High-pressure cylinder	(367·0 / 293·0) 74°	163·9 to 60	56·12	·45	206·7
Intermediate cylinder	(303·0 / 238·0) 65°	70 to 24	21·06	·375	209·5
Low-pressure cylinder	(238·0 / 134·0) 104°	24 to 2·59	11·34	·45	296·4
Condenser	130°	2·25			
Condenser supply ...	49·1°				
Air-pump discharge ...	64·6°				
Boiler feed	49·4°				
Totals				15·68 exp.	712·

TABLE XXV.—BALANCE-SHEET AS TO HEAT EXPENDITURE IN ENGINE PER INDICATED HORSE-POWER PER HOUR.

	British thermal units.	per cent.		British thermal units.	per cent.	
Heat supplied in pure steam in cylinder, due to temperature of saturation corresponding to pressure	15,905	99·6	Heat utilized by conversion into mechanical work	2,565	16·0	
Heat in ditto as superheat	none		Heat rejected : Heat from jacket-drains 17 Heat remaining in condensed steam ... ... 202 Heat acquired by condensing water 12,865	13,084	82·0 =	214·4 B.T.U. per minute
Heat supplied in steam in jackets	64	0·4	Heat radiated and unaccounted for	(320)	(2·0)	
Total	15,969	100·0	Total	15,969	100·0	

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CHAPTER XXXVII.

TOOTHED GEARING.

Strength of teeth.—The first requisite condition to be fulfilled by the teeth of a wheel is that they shall be sufficiently strong to bear in safety the load imposed upon them. Each tooth in a plain unshrouded toothed wheel sustains the pressure in a manner precisely the same as that in which an overhanging cantilever or corbel sustains its load. On the application of a load to a cantilever, in a direction at right angles to its length, the upper part tends to open or suffer extension, and the lower part to become compressed. When these conditions become sufficiently developed, the cantilever breaks. The weight which may be sustained before breakage is greater when the depth of the cantilever is increased. Just as a man instinctively spreads apart his legs or arms, to oppose more effective resistance to any pressure brought to bear upon him, so the cantilever will sustain a greater load if the parts which suffer extension and compression are separated, and one placed some distance above the other. But if the space be filled, or if in place of a divided beam a solid one of increased depth is substituted, two reasons exist whereby an increased load may be borne;

there is a greater area of material to oppose resistance to fracture, and the centres of resistance of the upper and lower portions are a greater distance apart. Hence the strength of such a cantilever increases as the square of the depth, so that one of double depth will carry four times the load, and one of treble depth will carry nine times the load. The load increases in exact proportion to the width of the cantilever, exactly as though two, three, or more separate cantilevers were placed side by side and separately loaded to the original amount. The possible breaking load diminishes in proportion to any increase in the projection or leverage, measured to the point of application of the load. These principles are strictly applicable to questions upon strength of teeth of wheels, which will be discussed at a later part of the present chapter.

Uniformity of pitch. — The second condition to be fulfilled by the teeth of a wheel is little, if any, less important than the first. This is that the teeth shall be placed around the circumference with the greatest possible degree of uniformity as to distance apart. Any tooth of which the front or working face stands in advance of its correct position, will take more than its share of the work, and will therefore be especially liable to break from the effect of fair stress alone. But an additional disadvantage exists in the fact that such abnormal pressure generally comes as a blow upon the tooth, which increases the chance of breakage, and which causes noise, either as a periodical thump, or as a rumble, or a roar, according to the degree of violence with which it occurs, and the frequency with which it is repeated. Any overloading of one or more teeth in a wheel must be accompanied by an underloading of others. Conversely, if it is found that any tooth or teeth are bearing lightly, it follows of necessity that the

work which they should perform is imposed as an additional load upon other teeth. In a wheel which has not been cut by an efficient machine, it very seldom occurs that the teeth come at once to such a bearing as to appear uniform. This irregularity necessitates greater strength in the teeth of such wheels than is necessary under a condition of greater uniformity. When the amount of irregularity reaches an appreciable amount, the repeated blows upon the teeth which are thus caused are almost certain to lead, sooner or later, to the fracture of the teeth, notwithstanding great excess in original strength. It is unnecessary to dilate upon the nuisance caused by the noise from a pair of heavy, defective wheels.

Manufacture of toothed wheels.—In current practice, the majority of wheels are put to work with the cast surfaces of their teeth in contact. Small spur- or bevel-wheels may be cast from whole patterns of iron or wood, or they may be cast in moulds, of which the teeth are formed separately by means of one block, which is adjusted in position for each successive tooth by dividing apparatus. In the case of larger spur-wheels, the teeth are cast upon segments, the pattern for which is built up in such a manner as to preserve its shape as correctly as possible, and the teeth are inserted and worked to shape, allowing for contraction. The construction of large toothed wheels has been described in the chapter on fly-wheels. Toothed wheels of moderate size are cast with rims, arms, and boss in one piece. The thickness of the rims is usually about equal to that of the teeth, and that of the arms is slightly greater near the rim, but increasing towards the boss, which is thicker still. The arms are best made of H-section, or with "face arms," which are continued as flanges along each side of the rim. Wheels of small and moderate sizes

are most efficiently made as plate wheels, with a continuous web, connecting the rim and the boss. If, however, the teeth are not properly made for true working, plate wheels will emit a loud ringing sound when driven at a high speed.

Inaccuracies arising in manufacture.—In each case, inaccuracies are liable to development. When iron patterns are employed, the castings may be found to be as perfect as the pattern. Some irregularity in ramming the sand may, however, slightly affect one or more teeth. But if such a pattern be machine cut, so as to give uniformity of spacing of teeth and smoothness of surface, to facilitate drawing from the sand, good castings are generally secured. Wood patterns are, however, subject to shrinkage and deflection, from which iron patterns are practically free. They change form by shrinkage or expansion, according as they are stored in a dry or damp place; and the castings follow the same general variation. If such alterations took place so as to leave the circular form of the wheel unaffected, the trouble would be sufficiently serious. But practically this is never the case; the arms, and the want of uniformity in the wood employed in the construction of the pattern, always give rise to distortion. Wood patterns are also liable to deflection under the strain set up in ramming the sand round them; this is especially liable to arise by irregularity in ramming between the teeth, so that large and small spaces are caused. Such irregularity is greater in castings made from old patterns which have become open or loose by shrinkage. A wood pattern should therefore be examined before use, as to circularity, and as to firmness of teeth; but especially as to defects which would be likely to cause irregularity of pitch. The most important condition is that the distance from the working face of one tooth to

that of the next one, measured along the pitch circle, is quite constant throughout the circumference. The thickness of tooth should also be uniform throughout, and the diagonal measure from the point of one tooth to the opposite point of the third or fourth, measured in each direction, should be also constant. In a wheel of which the rim is made up of a number of segments, any error which may exist in the pattern is repeated in each casting. An error, either constant or variable in amount, may also arise at each joint. When the mould for a casting has been made practically perfect, and a good casting run, it may become distorted if not allowed to cool with uniformity. The distortion in the teeth of heavy wheels is apt to prove very troublesome, in spite of all precautions. Some of the best brands of iron, in other respects, for wheel-making are especially liable to distortion.

Division and cutting of wheels by machinery.—The excessive distortion of wheels made in America led, many years ago, to the general use of machines for cutting the teeth of all important wheels accurately to form. Such machines had long previously been used in England for accurate work of small dimensions, and their use for heavy work is continuously extending. In such machines, the wheel to be cut is mounted on a spindle, which also carries a master-wheel of great accuracy. After the cutting of each tooth, and before commencing upon the next, the master-wheel, spindle, and wheel to be cut are revolved by suitable appliances through the proper distance to suit the required number of teeth. The amount of weight to be carried and the pressure of the cut to be resisted are very large, and therefore a very high degree of strength and rigidity in the machinery is necessary, for the attainment of fair geometrical truth in the work performed. These con-

ditions are opposed to the delicacy of adjustment which is necessary for good work. Many firms now practise the machine-cutting of large wheels, with good results. But in other cases, it must be admitted that much remains to be desired, chiefly in connection with the dividing part of the machine. In the dividing of light wheels of great precision, it is found to be impossible to rely upon the master-wheel. Upon the wheel to be cut, or efficiently connected with it, a circle is mounted, which is divided with great accuracy to the number of teeth required. A magnifying eye-piece, such as is adopted in surveying instruments, is used for observing the divisions upon the circle, and finally adjust the wheel for the cutting of each tooth. With proper precautions, one circle may be arranged to suffice for many numbers. The outfit need not be excessively costly, and the degree of accuracy to be secured in this way is of so much value, that such a means of division may be confidently anticipated to be ultimately considered to be essential for all good ordinary work. The care and attention bestowed upon wheels in this and other ways will naturally lead to an appreciable increase in original cost, but much lighter wheels will suffice for the work, while the efficiency and comparative silence which will be secured will be found to give ample return for the outlay.

Tools for cutting the teeth of wheels.—The majority of wheel-cutting machines are arranged for the use of milling-cutters. For cutting spur-wheels up to a moderately large size, the cutter is carried on a spindle whose centre stands tangentially to the wheel under operation. The milling-spindle is mounted on a carriage, which gives the cutting feed by sliding across the wheel, in the direction of the width of the wheel. But for wheels of a large pitch and of usual proportions of teeth

the adoption of this system of cutting would involve very large sizes of milling tools, which are difficult and costly to make in a solid form and to maintain, cause the imposition of heavy strains upon the work, and require exceedingly powerful machinery to drive them. In such cases a milling-block with separate tool-bits is used, or a milling-tool mounted on a spindle whose axis stands radially to the wheel under operation. The tendency in current practice towards a reduction in the length of tooth is clearly in favour of cutting of wheels, especially by machines of the first type. Compound tools are adjustable and less costly than the original type, and any portion of the cutting edges which may suffer damage can be conveniently renewed. After commencing the finishing cut upon a wheel, the outline of the cutter should not be altered by grinding until the wheel is finished. Measures should also be taken to avoid or compensate for irregular or local heating of the work.

Machines are made for wheel-cutting by means of a tool of the same class as used on a shaping machine. The tool is carried on a sliding ram, moving in a carriage which is guided in accordance with the form of a metal template, shaped to suit each wheel. By this means the necessity for very high power to drive the tool is avoided, the machine is less costly, and a less costly stock of cutting tools is required. This type of machine can be arranged to cut bevel-wheels quite correctly, for which purpose it is of greatest value.

Teeth to be cut on all surfaces.—The teeth of all machine-cut wheels should be finished on the points and ends, so that every surface is treated, whether it comes into actual contact with its fellow wheel or not. By this means the accuracy of the whole becomes more directly secured and completely verified.

Uniformity of transmission of motion.—But though accuracy of division and cutting will in every case lead to important improvement, and give the means whereby to avoid many defects to which wheels are subject, yet, unless the working face of each tooth is correctly formed, a satisfactory result is impossible. An ideal pair of wheels is presented, when two discs, whose edges form smooth cylindrical surfaces, are placed in contact, and one is revolved in such a manner as to drive the other by mere frictional contact. With perfect driving in this manner the length of surface traversed in a given time is the same upon each disc. The edges of such discs are shown in Fig. 101, in which the point of contact is defined by 100. Points 93 to 107 are obtained by measuring equal lengths—not chords, but arcs—along the curve of the larger wheel, in each direction from 100. In like manner, points 93*a* to 107*a* are spaced along the circumference of the smaller wheel, each being equal to each of the spaces upon the larger wheel. When the two wheels are revolved in contact without slipping, the corresponding points on each will successively and precisely meet at the point of contact of the wheel. The principle of transmission by rolling contact of smooth surfaces is largely adopted in friction hoists, owing to special convenience in making and breaking contact. But, as is well known,

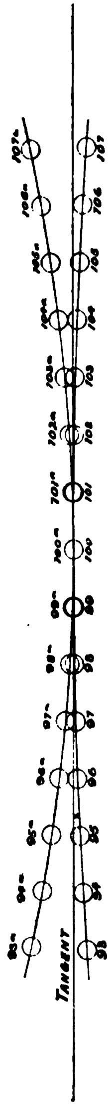


Fig. 101.—Discs for transmission of motion by frictional contact of smooth surfaces.

slipping takes place, the power of the arrangement is strictly limited, and excessive pressure is imposed upon the bearings, in order to maintain effective contact. But by breaking up each surface into a number of teeth, so that the teeth of one wheel fit into the spaces of the other one, slipping becomes impossible. There is an unlimited choice in the forms of teeth, which would amply suffice to provide original strength and prevent slipping. But it is essential to success that the transmission of motion should be effected with the same uniformity as in the case of the smooth discs, and this condition imposes very narrow limits upon the choice of forms of teeth.

In considering any question as to design of the teeth of wheels, each wheel is, in the first instance, taken as a smooth cylindrical object, of which the surface is called the pitch surface. It is represented on paper by a circle, and is hence usually referred to as the pitch-circle, but the constant basis of discussion is the cylindrical surface represented by the circle. These circles correspond to the two of which portions are shown in Fig. 101, and if the teeth are correctly formed, the circles will roll in contact without slipping, and points of division, as in Fig. 101, will accurately correspond with each other during revolution, while one or more teeth of one wheel will always be in contact with those of the other wheel. This principle is applied in the most practicable and absolute check upon the correctness of design of wheel teeth, and which is dealt with at a later part of this chapter.

A complete discussion of the several forms of wheel-teeth cannot be here entered upon. But the two forms which entirely fulfil the condition of uniform transmission are the involute and the cycloidal forms. Involute teeth cause the imposition of an excessive

thrust upon the bearings, and are seldom found to be necessary or desirable, except for use in rolling mills and other like situations, where the distance apart of the centres requires to be variable. Teeth of the cycloidal type are therefore used in ordinary work to the practical exclusion of all others.

Cycloidal teeth.—A true cycloid is described by a point in the circumference of a circle, rolled along a stationary straight line. An epicycloid is described by a point in the circumference of a circle which is rolled in contact outside another circle. A hypocycloid is described by a point in the circumference of a circle which is rolled inside a larger one. The form of the teeth of a straight rack is based upon the cycloid; the faces or points of the teeth of wheels upon the epicycloid; and the flanks or roots of the teeth of wheels upon the hypocycloid. In each case those portions of opposing wheels which come into contact with each other must be described by rolling circles of equal diameter; and in each case the rolling circle is applied to the pitch circle of the particular wheel. There is a fairly wide choice as to the radius of the rolling circle to be adopted. As a rule, it is convenient to assign the same rolling circle to the teeth of all wheels of same pitch, so as to give facility for interchanging. If the radius of the rolling circle is less than half of the radius of pitch surface of smallest wheel which may possibly be made, it will give teeth of a very weak form in the root. With this point in view, Professor Willis proposed that the radius of the rolling circle should be one-half that of the pitch-surface of the wheel of twelve teeth, which would thus become the minimum wheel of the set. The radius of the rolling circle which agrees with this condition is found by multiplying the pitch by ·955. Independently of any reference to wheels of

small diameter, good results have been secured by the use of a rolling circle whose radius is equal to the pitch of the teeth. For ordinary work, there appears to be no good reason for the introduction of fractions for this purpose, especially as the result can only be affected to a minute extent. But under the system of Professor Willis, the direct use of the cycloidal curves is superseded by that of circular arcs, the radii of which are derived from tabular numbers, so that its application would not be facilitated by the adoption of a slightly different rolling circle. A rolling circle of smaller radius will give a tooth of stronger form, but upon which the thrust is applied with a greater degree of obliquity. The teeth will also remain in contact for a shorter time, or the "arc of contact" is shorter. The converse conditions apply to an increase in the radius of rolling circle. The length of the arc of contact, or the distance through which the teeth will move without interruption of contact, is defined by the application of the rolling circle to the point of contact in profile. The intersection of each outer rolling circle with the circle of the points of teeth of its own wheel gives the length required, and as shown upon each wheel in Fig. 102. The process of setting out a pair of wheels, by epi- and hypocycloids, is a somewhat troublesome one. But every pair of moderately large wheels is quite worthy of this trouble. When the two curves have been obtained, circular arcs closely approximating to them should be found, for convenient use in the workshop. This subject is treated most exhaustively in many works, in most of which are given rules for approximate cycloidal teeth, produced directly by the use of circular arcs, and of various degrees of merit. Most of these give one arc for the face, and another one for the flank of the teeth of each wheel. But a few give two arcs

for the face and two for the flank. An actual study of the

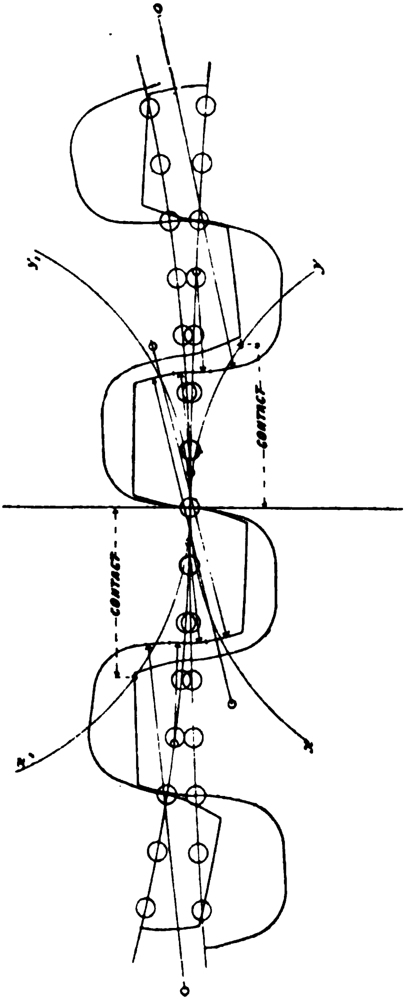


Fig. 102.—Teeth of wheel, rolling in contact.

true curve will show the desirability of giving two arcs to each part. Notwithstanding, however, the exercise of

the greatest skill and care in designing the shape of the teeth, the check by rolling should not be omitted.

Testing profile of wheel teeth.—Very accurate tracings are made to show the outline of several teeth in each wheel, with the pitch-line of each clearly marked. These tracings should be kept quite smooth and unfolded until they are no longer required. Short equal spaces—say one-fifth of the pitch—are stepped along each pitch-line, as in Fig. 102, in which the two tracings are assumed to be laid in contact. Around each of the points so determined a small circle is drawn, which may very conveniently be arranged of such a diameter that the opposite ones, at say the fourth division from the point of contact, will just touch each other. The pitch circles may be brought into contact at the successive points 93 . . . . . 107, as in Fig. 101, with the division marks exactly coincident at the point of contact, and the tangential position defined by the contact of the small circles at the fourth point from the point of contact. The length of the arc of the pitch circle covered by actual contact of the teeth is thus shown. If in the first trial the teeth are found to overlap or to fail to make contact, the form should be modified, and a second trial made. In the example shown in Fig. 102 practically true contact is maintained in seven positions, or over six intervals, or $1{\cdot}2 \times$ pitch. At all times, therefore, one tooth of each wheel is in true contact with a tooth of the opposite wheel, while during a considerable portion of the time two pairs are in contact.

Teeth of wheel of imperfect profile.—As a contrast to the above, an example is given in Fig. 103 which shows such a design as would have been determined upon by many millwrights a few years ago, under a mistaken impression that the face and flank of a wheel tooth

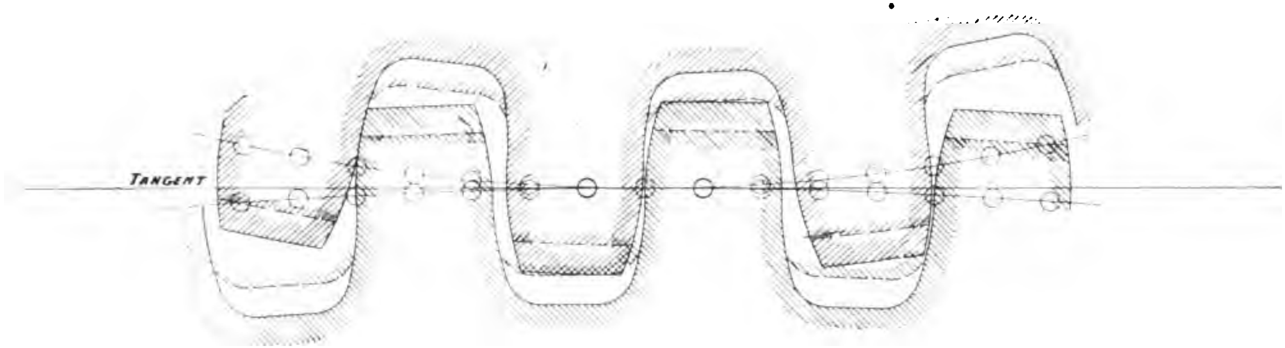


Fig. 103.—Teeth of wheel, of imperfect form.

should be struck to a radius equal to the pitch, and that the centre of each arc, whether for the face or the flank, should be situated upon the pitch circle. This resulted in making the teeth too full in the points, and the points of the teeth of the larger wheel always carried more than a fair share of the pressure, in fact practically the whole of it. Contact took place at some distance from the line joining the centres of the wheels, so that the amount of sliding which occurred between the teeth, while under pressure, reached its maximum; and consequently also the amount of loss by friction. In all such cases this loss was excessive, though slightly less when the large wheel was driving than in the opposite, because in the former case the pressure was on that side which was retreating from the line joining the centres of the two wheels. The fact that contact is persistently made at the points of the teeth places such teeth at a serious disadvantage in point of strength. When both wheels are revolving at uniform velocity, one tooth can only remain in actual contact with its opposing tooth over about one-third of the length of pitch. If under such conditions uniformity of motion could occur, it would follow that one wheel would drive the other for only one-third the time, and that during that period the actual pressure upon the teeth would be thrice as great as the apparent or mean pressure. As a matter of fact, however, such wheels do not transmit uniform motion; one pair of teeth are receding or allowing a reduction in velocity, while the opposite condition obtains with the next pair. The second pair therefore come into momentary contact to the accompaniment of a distinct blow, the motion is transmitted in a series of jerks, and noise is produced. At ordinary speeds the separate blows cannot be distinguished by the ear; but this becomes possible when they move

at a low speed, as in stopping and starting the engine.

One means largely adopted, with a view to improve this form of tooth, was to draw a rounding line at some distance below the pitch-line, upon which line were placed the centres from which the faces of the teeth were struck. The flanks were still struck from points upon the pitch-line. In each case, the radius remained as before equal to the pitch of the teeth. By this means the points of the teeth were somewhat relieved; but still the teeth were far from correct, and the only true remedy is the adoption of cycloidal curves for the teeth. Willis's and Adcock's forms of teeth are much superior to the old forms described, and have perhaps been more used than any others, but many others have also been adopted.

Pitch to be measured along circumference.—In all wheels the pitch should be measured along the curve of the pitch-line, so that when correctly drawn, the distance from centre to centre of the teeth, measured directly along the chord, is not precisely equal in corresponding wheels of different diameters. But if a pair of wheels are set out on equal chords and teeth assigned, as shown in Fig. 103, they will be found to fit slightly better than if designed with the pitch measured circumferentially. It is, however, impossible to regard this otherwise than as an attempt to correct one error by introducing another.

Correctness of profile as evidenced by uniform contact. Wheels which are correctly designed, made, and fitted will be found to bear evidence of *uniform* contact from the point of each tooth to some distance below the pitch-line, which distance cannot be generally defined. Every tooth should show these signs, and until this is the case, the wheels must be a nuisance, and the

cause of more or less trouble. They will cause excessive loss of power; they will be placed at a disadvantage in various respects as to strength, they will be unnecessarily noisy, and the jerky motion which they transmit is far less efficient than a smoother motion. One tooth, incorrectly placed, may suffice to cause breakage.

Rectification of imperfect wheels.—Wheels which are found to be very slightly imperfect when first put to work, may be allowed to remain untouched until they wear down to a good surface. As a rule, however, this is a costly proceeding, and all hard-bearing places should be thoroughly chipped over and in some cases roughly filed. This may have to be repeated two or three times before a satisfactory result is secured. Grease should be sparingly applied until the operation is completed. The work is tedious, and has often to be performed in cramped and dirty situations. Contractors, workmen, and owners are therefore easily persuaded that chipping and filing are unnecessary or injurious, and that the hard skin of cast-iron should be scrupulously left intact. When, however, the importance of the office fulfilled by the teeth of a wheel is fully considered, it will be found that to leave them with black surfaces of incorrect shape is just as unreasonable as the same would be in connection with cross-head slides or shafting bearings.

Clearance between teeth of wheels.—Under certain ideal conditions, it may be possible to make a pair of wheels which shall always be in contact on both the working faces and the backs. But under practical conditions, every pair of wheels requires to be made with a certain amount of clearance at the back of the tooth, which in very few cases should be less than one-fiftieth part of the pitch. One thirty-second part, as in Fig. 102, is usually better; and for wheels of small pitch

this fraction may be largely exceeded. When this is made very small, there is greater liability to damage, by reason of cross bearing of teeth, or by reason of small objects which may drop into the wheels when at work. The liability to breakage by backlash may be increased or reduced. But the occurrence of backlash to any serious extent should be avoided by ensuring a sufficient degree of uniformity in turning. On account of the back clearance, the thickness of the tooth (T, Fig. 104) requires to be reduced below one-half of the pitch, in at least one wheel of each pair.

Length of teeth.—The length of the tooth is an element of great importance, as affecting the strength.

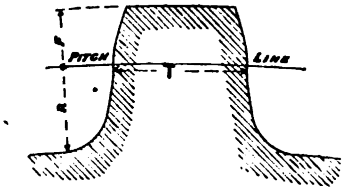


Fig. 104.—Diagram of wheel-tooth, as to proportions.

The total length is divided into *F*, the face length, and *R*, the root length, as in Fig. 104. Common proportions for these are respectively ·3 and ·4 of the pitch, but they are much more appropriately made ·2 and ·3 of the pitch. The latter proportions were given by Adcock in *The Engineer* of 17th Sept., 1869, and have been largely and successfully used since that time. The bottom clearance need not be so great as one-tenth of the pitch in every case, but in every case it should be sufficiently great to allow an ample fillet to be provided at the springing of the tooth, which is a point much too frequently overlooked. Still shorter teeth have been advocated, but very few cases arise which cannot be fully met by Adcock's proportions. Long teeth remain

in contact over a longer arc than do short teeth. Teeth of good design, as in Fig. 102, are always in fair contact at one place, if made of the length recommended. It will be found impossible by means of any practicable increase in length to provide that two pairs shall always be in contact. Therefore there is but little advantage to be secured in this respect by an increase in length.

Factor of safety, or reserve strength.—From the foregoing account, it is clear that in every step in the designing of wheels, attention should be paid to every possible means towards ensuring absolute freedom from shock. But when every precaution has been taken, it is almost impossible to say that accidental causes will not at some time interfere with perfect working, and minor vibration arise. Even if the teeth themselves and the attendant conditions are so marvellously perfect that shock and vibration are completely absent, the stress due to the work is applied at one moment and is removed the next. It is proved by experiment and by long experience that a load applied to a structure and many times repeated will cause breakage, even though the load may be much smaller than would have sufficed to break it at one operation. In one case it may be necessary to apply the load 1000 times, in another 1,000,000, and in another 1,000,000,000, according to the magnitude of the force applied and the manner of operation. The structure must therefore be made several times stronger than would directly appear to be necessary. This ratio is called the factor of safety, and in toothed wheels is a high one.

Teeth of the type shown in Fig. 102, made of good cast-iron and loaded in accordance with Table XXVI., possess a factor of safety of about 32, assuming that the load on the tooth does not at any time exceed the

average load due to the work, and also that such a load is applied to the extreme end of the face of the tooth. But the full-length tooth shown in Fig. 103 only possesses two-thirds of the strength of the tooth shown in Fig. 102. The actual strength may, however, be increased by shortening the tooth, as shown in dotted lines, so as to reach within about 8 per cent. of that of the tooth shown in Fig. 102, which is of equal length. But the irregularity of load to which such wheels are exposed in work reduces the factor of safety to a variable but very great extent.

Table relating to strength of wheels.—The power which may be transmitted by spur-wheels of cast-iron moderately well made, carefully erected, and in good condition, designed as in Fig. 102, may be obtained from Table XXVI., which is calculated upon a width of tooth equal to two and a half times the pitch. The indicated horse-power value is given for each pitch. This, multiplied by the number of teeth in the wheel, and by the number of revolutions per minute, gives the indicated horse-power which the wheel will transmit. The vertical columns give the numbers of teeth in wheels of different pitches, which are equal in strength to the shaft whose diameter is given at the head of each column. This table is based upon an admissible pressure due to the work done of 317 pounds for a wheel of 1 inch pitch, $2\frac{1}{2}$ inches in width. The pressure upon wheels of other pitches, and of proportionate width, vary according to the square of the pitch, in conformity with the principles referred to at the beginning of the chapter.

Rules as to power of wheels.—A rule very much used in Lancashire gives the indicated horse-power =

$$\frac{\text{Pitch}^2 \times \text{width in inches} \times \text{velocity in feet per second}}{16}.$$

TABLE XXVI.—STRENGTHS OF WHEELS AND SHAFTS.

Wheels of cast-iron, uncut.			Diameters of iron shafts in inches.																
			2	2¼	2½	2¾	3	3¼	3½	3¾	4	4½	5	5½	6	7	8	9	10
Pitch.	Pressure.	I.H.P. values.	Indicated horse-power values for shafts, to be multiplied by the number of revolutions per minute.																
			·099	·141	·193	·257	·383	·424	·529	·651	·790	1·125	1·543	2·054	2·667	4·234	6·321	9·000	12·346
inches.	pounds.	×T×R	Numbers of teeth in equivalent cast-iron wheels.																
1	317	·0008	124	176															
1¼	495	·0016	63	90	124	165													
1½	713	·0027	37	52	72	95	124	157	196										
1¾	970	·0043	23	33	45	60	78	99	124	152	184								
2	1,268	·0064	16	22	30	40	52	66	83	102	124	176							
2¼	1,604	·0091	11	16	21	28	37	46	58	72	87	124	169						
2½	1,980	·0125	...	11	16	21	27	34	42	52	63	90	124	165					
2¾	2,396	·0166	...	...	12	15	20	26	32	39	48	68	93	124	160				
3	2,853	·0216	...	...	...	12	16	20	25	30	37	52	72	95	124	196			
3¼	3,348	·0275	...	...	...	...	12	16	19	24	29	41	56	75	97	154			
3½	3,882	·0343	...	...	...	...	...	12	16	19	23	33	45	60	78	124	184		
4	5,072	·0512	...	...	...	...	...	...	...	13	16	22	30	40	52	83	124	176	
4½	6,417	·0729	...	...	...	...	...	...	...	...	...	16	21	28	37	58	87	124	170
5	7,925	·1000	...	...	...	...	...	...	...	...	...	...	...	21	27	42	63	90	124
6	11,414	·1728	...	...	...	...	...	...	...	...	...	...	...	...	16	25	37	52	72

Iron cut wheels, or steel wheels carefully trimmed, are 25 per cent. stronger than table. Numbers of teeth of such wheels as per table correspond to steel shafts of diameter given. Strength of steel cut wheels is 50 per cent. greater than given in table.

This rule agrees with Table XXVI. when the pitch of the wheel is about 3·7 inches. It is generally agreed that this rule gives light proportions with large pitches, and comparatively heavy wheels for light work. Progress has been made during recent years, and wheel teeth are made of better form and proportions, so that the strength and uniformity of loading have been improved, and a much heavier load may be now imposed. For this reason the power which any wheel will transmit is given in Table XXVI. as 9 per cent. greater than the similar table given in 1879. This improvement may be reasonably expected to continue, so that still heavier loads may be applied. When cut cast-iron wheels are adopted, an addition of 25 per cent. may be made to the power given in the table. Cut wheels of cast-steel are 50 per cent. stronger than given in table. Uncut steel wheels should not be used without careful trimming When this is done the strength may be taken as equal to that of cut cast-iron wheels, or 25 per cent. above the strength given in table. The teeth shown in Fig. 102 are 6 inches pitch, and assumed to be 15 inches in width. Under favourable conditions such wheels, made in iron and driven at a speed of 2,313 feet per minute, would transmit 800 indicated horse-power, or 1,200 horse-power if well made and cut in steel.

Means adopted for the purpose of increasing power.—Various means have been adopted with a view to increase the power which may be transmitted by a pair of wheels of given proportions and pitch. In addition to the power to carry an increased load which is imparted by improved form and proportions already dealt with, the following are the chief means adopted :—Increase of speed, increase in width of wheel, shrouding, helical disposition of teeth, and buttress teeth.

Working velocity of wheels.—A high speed for

toothed gearing in ordinary practice is about 2,500 feet per minute. If the teeth are at all inclined to be noisy, they become quite intolerable at any higher speed, though well-designed wheels may be so driven. One of the earliest instances of this—and a most successful one—was exhibited by Mr. Corliss at Philadelphia in 1876, in which a spur-wheel of 30 feet diameter was driven at 36 revolutions per minute, giving a gearing speed of 3,402 feet per minute. These wheels were machine cut, of 5¼ inches pitch and 24 inches wide, and most undoubtedly they worked with exceeding quietness. At the time of the Exhibition they transmitted 1,400 indicated horse-power, but were intended to transmit 2,500. The latter amount will be found to be almost in accordance with Table XXVI., if proportionate allowance is made for the exceptional width, and an addition of 50 per cent. made, instead of 25 per cent., as given above for cast-iron cut wheels.

Special provision for strength, necessary at high velocities.—In all cases in which wheels are driven at a high speed, care must be taken to provide sufficient strength in the fly-wheel to resist centrifugal force.

Width of wheel along teeth.—An increase in the width of the wheel or the transverse length of the tooth while the pitch and form of tooth remain constant, increases the strength of the tooth under fair loading in exact proportion to the width of the wheel. But it often happens that one shaft is allowed to wear down at one end, so that the bearing of the teeth is affected, the load thrown very much to one side, and the advantage of a wide wheel entirely lost. In some extreme cases, where small clearance was originally allowed, the wheel teeth bear on opposite corners, causing the imposition of such a strain as renders the failure of the teeth quite certain Such a defect may become developed by reason of a

settlement in the structure or foundation of the work, or by wear of bearings. Only in cases in which the probabilities in this direction are remote is the adoption of wide wheels to be recommended. When the shaft thus affected is a short one, so that the bearings are comparatively close together, a settlement of definite amount will cause a greater disturbance in gearing than would otherwise be the case. This is true under all conditions, but the character of the derangement is different according to the position of the point of contact between the wheels with respect to the vertical plane through the shaft. When there is a reasonable probability of maintaining good contact across the teeth, an increase in the width is beneficial in reducing the load per inch of width, apart from the strength of the teeth. Sir William Fairbairn recommended that the pressure should never exceed 400 pounds per inch of width. This condition should still be observed if possible, but in many cases higher pressures must be adopted. For uncut teeth of very good design in cast-iron 800 pounds per inch may be allowed, when kept well lubricated. Steel wheels may be loaded in the proportions already given. Under favourable conditions, the width of wheels may be allowed to reach three to three and a half times the pitch.

Shrouding or flanging of wheels.—Shrouding consists in the provision of a flange or connecting-piece, cast solid with the teeth. When both wheels in a pair are shrouded, the shrouds can only reach up to the pitch-line, or say half-way up the teeth. When only one wheel is shrouded, the shrouds may reach the tops of the teeth. In this case, the smaller wheel is generally the one taken for the purpose, owing to the more frequent contact of the teeth, and consequent increased wear. Shrouds are also generally more conveniently

applied to the smaller wheel. But when the larger wheel is of iron, and the smaller one of steel, the wear in the former is the greater, and thus the shrouding is sometimes applied to the larger wheel. When wheels are exposed to unavoidable shocks in work, so that it is practically impossible to provide strength to give a sufficient margin for safety under all conditions, teeth are liable to be broken out. It is less inconvenient that this should occur to the pinion than to the wheel, and for this reason the teeth of the latter are sometimes reinforced by shrouds, while the former are left plain. Other measures for the same purpose consist in making the teeth of different thicknesses, and in the provision of frictional slipping or safety centres to one or more wheels. In all cases, it is practically impossible to cut by machine the teeth of a shrouded wheel. The operation is just possible with a radial milling-cutter, but the recessing at each end of the cut interferes with the value of the shrouding, though some still prefer to apply shrouds, chiefly for the sake of appearance.

Helical teeth.—The use of helical teeth is largely practised. It has already been shown that teeth whose outline is correctly designed will transmit motion with its original degree of uniformity, while teeth which are incorrectly formed will transmute a steady motion into an irregular or jerky one. This defect has been sought to be overcome by means of the well-known stepped wheels, which are used for many purposes in which a pinion is required of diameter smaller than is possible in a plain pinion of pitch sufficient for strength. For wheels of any ordinary diameter such stepping is quite unnecessary, if the teeth be properly designed. If the outline is not correct, stepping cannot affect the result, except by causing the imposition of the entire load

upon one step after another, which causes overloading of small portions of the teeth in succession, and leads to rapid wear if not to breakage. Simple helical wheels are a further development of stepped wheels, which introduce an objectionable end thrust upon the shafts. Double helical wheels, which are largely used, are designed with opposing helices to balance the end thrust arising from the use of simple helical wheels. In both single and double helical wheels the pressure due to work transmitted is met obliquely, and is therefore greater in amount than it would be if met directly, as in an ordinary spur-wheel. But the thickness of tooth in a direction normal to the pressure is less than in a spur-wheel of equal pitch. The transverse length of tooth, or the width of wheel, measured along the line of tooth, is correspondingly greater in the helical wheel, but the nett strength is certainly not greater, though the pressure to be overcome is so. But if helical wheels are of value in the manner ordinarily claimed for them, they must necessarily be loaded over part of the width of the wheel, and this is quite unavoidable unless the marking of the teeth shows them to bear over a large area of face. Otherwise—and in any ordinary case—each point in succession is still further overloaded. They cannot be subjected to ordinary machine cutting. They are apt to cause scoring of bearings by reason of the utter absence of end play. In short, they have quite failed to display any advantages which cannot be more perfectly secured by other means.

Buttress teeth.—When a pair of wheels are required to run only in one direction, and are never subjected to backlash or pressure on the backs of the teeth, these need not be made to the same shape as the working faces. In this manner the root of the tooth can be very largely strengthened in a convenient manner, as in

Fig. 105, which is reproduced from *The Engineer's and Machinist's Assistant*, 1847. This practice has been revived and largely followed during recent years. When a large pair of such wheels are subjected to severe backlash, the wedging action of the backs of the teeth causes the imposition of severe stresses upon the shafting, on the wheel teeth and the rims. This is reduced by relieving the obliquity, so that the shape of the back may more nearly approach that of the face. Probably in no case is it wise to carry the principle to its full development.

It does not appear that any of the measures just described present any advantages which cannot be at

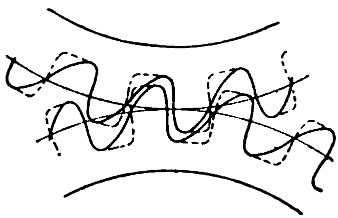


Fig. 105.—Buttress teeth of wheels.

least equally well secured by the adoption of plain teeth well proportioned, and accurately designed and constructed.

Mortice-wheels.—Mortice-wheels are now comparatively little used. Their office is to provide a medium of greater elasticity than iron, to yield under shocks, whether arising from the defective form of the teeth, or from any extraneous cause. Sometimes they are used for export to places where a fully skilled millwright is not available for their erection, and where the noise from common iron wheels is an objection. A second, but very important, reason for their comparative silence consists in the fact that the wooden teeth will wear to

relieve their own asperities or those of the fellow iron wheel, while the asperities of a pair of iron wheels, if not chipped or filed away, will remain for many years and perhaps cause breakage. The useful strength of well-designed mortice-wheels is about the same as that of plain iron ones of equal pitch. The wooden tooth is thicker and the corresponding iron tooth is rather thinner than an ordinary iron tooth. The clearance may be safely made less than with iron teeth. When mortice-wheels are adopted, the factor of safety of the teeth of both wheels is much less than with ordinary iron wheels, but the elasticity of the wood renders a high one unnecessary. There is, however, very seldom any real necessity for the adoption of mortice-wheels.

Bevel-wheels.—Bevel-wheels are based upon rolling conical surfaces, in the same way that spur-wheels are based upon rolling cylindrical surfaces. The general conditions as to shape of teeth apply equally to spur-wheels and bevel-wheels, and a tooth which, from its shape and proportions, would be suitable for one purpose, would be equally suitable for the other. The form of a tooth of a bevel-wheel is not taken in a plane perpendicular to the centre of the shaft, but to the pitch surface, which lies nearly along the centre of the tooth.

The power which may be transmitted by bevel-wheels may be ascertained from Table XXVI., in the same way as may that of spur-wheels, the pitches and diameters being taken at the large ends of the teeth in the usual manner. By this means the bevel-wheels receive a rather lower factor of safety than do spur-wheels. But on the whole, bevel-wheels are rather more easily kept in efficient order than spur-wheels. In a pair of ordinary bevel-wheels, the sum of whose teeth is 100, the actual strength of the tooth is about 20 per cent. less than in

the case of a spur-wheel of equal pitch in equal condition, while with larger wheels the difference is less.

Bevel-wheels are more difficult and costly to cut accurately than spur-wheels. As they are generally of smaller dimensions than spur-wheels, there is less difficulty in getting them to work satisfactorily in an uncut condition. In all cases where one tooth bears harder than another, it should be chipped or filed to an even bearing. If a large or small number show a tendency to bear hard at the points, and to avoid bearing lower down, they must also be eased so far as to overcome such tendency, or relieved by adjustment of shaft. The importance of patient attention to this operation, so as to secure silence and efficiency in work, has been already referred to in connection with spur-wheels. Here also the work should not be postponed until the wheels and their surroundings are so thickly coated with grease that the work becomes quite unnecessarily difficult and disagreeable.

CHAPTER XXXVIII.

BELT GEARING.

Adhesion of belting.—Belting is employed for the transmission of motion from one revolving shaft to another, by means of frictional adhesion upon pulleys with approximately cylindrical faces. The surfaces of both belt and pulley should be in the condition best adapted to secure efficient and intimate contact. Both surfaces should be clean and free from loose matter of all kinds, whether in the form of a continuous film or in patches. The surfaces of the pulleys should be accurately turned, dead smooth, but not to a high polish. It is sometimes considered that a slight roughness upon the turned surface is an advantage, in improving the hold of the belt, but this is not the case. Roughness also tends to cause the accumulation of dust and dirt, which forms cakes of irregular thickness, and leads to useless wear of belts. When a belt has been at work some time and the pulleys assume a brightly-polished condition, it is a proof that slipping occurs. A dull polish is equally a sign of good work. As the belt wraps round the pulley, the air should be squeezed out completely, to allow contact between the surfaces. If this is prevented by any cause, whether by roughness

of surface of pulley or the temporary occurrence of collections of dirt, the belt fails to hold, and slipping occurs. Holes made in the belt or the pulley are beneficial in allowing the free escape of air. Such holes should be made of elongated form, with the greatest dimension placed longitudinally, so as to interfere as little as possible with the strength of the belt or pulley. A hard unyielding belt is much inferior to a soft supple belt in respect to efficiency of contact. The former also causes loss of power in bending the belt round the pulleys, and in restoring it to a straight condition on leaving.

Tensions upon belt, on account of work done and of velocity.—The frictional adhesion of a belt upon a pulley prevails by reason of pressure between the surfaces, which pressure arises from the pull of the belt. If it were possible to work a belt with absolutely no tension upon the slack side, the tension in pounds upon the tight side, multiplied by the velocity of the belt in feet per minute, would give the number of foot-pounds of work done; this being divided by 33,000 would give the actual horse-power transmitted. But even when the surfaces are in the best condition, it is found that the tension upon the slack side must be about one-third of that on the tight side, while it usually reaches one-half. The maximum total tension upon a belt or that upon the tight side, may be divided into three parts. The first part is the tension or stress due to the work absolutely transmitted from one shaft to the other, and may be termed the primary stress. The second is that necessary to maintain sufficiently tight contact and prevent slipping; this may be termed the secondary stress. The third part is that due to centrifugal force. The slack side is subject only to stresses of the second and third orders. The ratio between the

primary and secondary stresses depends only upon the condition of the surfaces, and is quite independent of centrifugal force. The secondary stress and centrifugal force on the two parts balance each other; by means of the former a very considerable pressure is imposed transversely upon the bearings; but the latter has no effect in this respect.

Pressure upon bearings due to tensions of belt.—As a rule, the pressure upon the bearings of a shaft, due to the use of a belt, may be taken as the sum of the primary stress upon one side, and the secondary stresses upon both sides. Usually, these are approximately equal to each other, so that the sum equals three times the primary stress, or that due to the work done. But in cases where the two pulleys are of widely different diameters, this is not strictly correct. An elementary application of the principle of the parallelogram of forces will, however, show that no important error is thus introduced. The *total* amount of pressure upon the shaft bearings may be found in a similar way, by combining the stress due to the pull of the belt with the weight of the shaft and fittings acting vertically downwards.

Choice of surfaces of belt for contact.—A new belt will work most easily with the smooth or grain side in contact with the pulleys. But if the rough or flesh side of the belt is applied to the pulleys, it will gradually acquire as good a surface for holding as that of the grain side. At all times the belt will be found to possess greater softness, elasticity, and durability if applied with the flesh side towards the pulleys.

Creeping of belt.—All belts which work upon smooth pulleys must of necessity "creep" or "slip" upon the driving and driven pulleys. This is evident from the fact that the belt suffers less stress upon the slack side

than on the tight side, and therefore it suffers less stretch or extension in length at one time than another. For this reason, a longer length of belt is delivered from the driven pulley and received on the driving pulley than is delivered from the driving pulley or received on the driven pulley. The difference in length thus arising is made up by slip on the pulleys themselves. The practical effect of this action is to cause a slight loss of speed in the driven pulley. The application of greater force in stretching belts when applying to the pulleys causes an increase in the secondary stress, and a reduction in the amount of creep. Such a measure, however, leads to greater pressure upon the bearings, and greater wear and tear of the whole plant.

Transverse convexity of pulleys.—Usually, pulleys are made convex in cross-section, for a reason to be subsequently referred to. For this purpose an easy curve should be adopted, with a view to avoid injury to the belt. An arc of a circle is more convenient than any other curve in the design and execution of the work, and is usually adopted. Excessive rounding of the pulley face causes a certain amount of slipping, by reason of a potential difference in the velocity of the periphery, at the centre of the belt and at its edges. No absolute difference in the velocity of the several parts of the belt is possible, as the whole moves together. Hence it is obvious that the difference in velocity must be covered by slip. If such a curve be divided into strips, so that the difference between the largest and the smallest diameter in each strip is one-quarter of the total difference in diameter between the centre and the edges of the belt, it will be found that each half width is divided into 25, 10½, 8, and 6½ per cent. of the total width, the narrow strips being at the edges. Fig. 106 shows such a curve, much exaggerated for the sake

of distinctness. An appropriate curve may be obtained by rule—

$$\text{Versine of belt in inches} = \frac{\sqrt{\text{width over belt in ft.}}}{8} = \frac{\sqrt{\text{width over belt in ins.}}}{27{\cdot}7}.$$

This should, however, be never less than one-eighth of an inch. When a belt is running without load, the centre parts overpower the narrower and less powerful parts on each side, and the belt slips at the edges, an additional reason for this being that the belt is naturally slacker by a very small amount at the edges. If, however, the belt is heavily loaded, the load tends to make the belt slip at all points. Usually the amount of this slip is very small, but such a tendency is largely

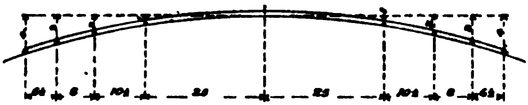


Fig. 106.—Transverse rounding of pulley face: exaggerated.

promoted by the adoption of excessive rounding. If it should attain such proportions as to cause heating of the pulley rim, or the production of a highly-polished surface, immediate measures should be adopted for prevention. Such a condition causes direct waste of power, in proportion to its amount; it leads to irregularity and uncertainty in transmission; it leads to the premature destruction of all belts subject to it, and liability of the belts to run off the pulleys, causing at least a stoppage of work.

Leather surfaces to increase adhesion.—The slipping of belts may be sometimes prevented by the adoption of a covering of leather upon the pulleys, which gives a higher co-efficient of friction than does bare iron.

Flanges to prevent belts running off the pulleys.—These are sometimes adopted, but usually they are practi-

cally powerless for the purpose, especially in connection with wide belts. If, notwithstanding the flanges, the belt does run off a pulley so provided, it is certain to be cut. For ordinary work they are therefore quite useless, but they are very convenient in a few cases in which a belt is required to run slack upon a pulley, and tightened by a movable pulley, when very little restraint is required to prevent it from running off.

Overloading of edges to be avoided.—It is sometimes proposed to cut a belt rather shorter at the edges than in the centre, to correspond with the rounding of the pulley. But if everything is in good order, there is no necessity for any such measure, and in any case it is better to leave the edges rather underloaded.

Transverse rounding of pulleys mounted upon vertical shafts.—The rule just given allows a very suitable amount of rounding for pulleys which work upon horizontal shafts. For the few cases in which vertical shafts are employed, the rounding should be at least twice as great, or thrice if working at a high speed. In such cases the additional rounding may with advantage be made rather greater in the lower half of the pulley; but when this is adopted, the pulley so treated should receive special attention, with a view to prevent its inverted application.

Belting calculations.—The chief factors entering into calculations upon belting are—the power to be transmitted, the tensile strength of the belt, the proportion of tension which may be required to be applied to the slack side of the belt, the velocity at which the belt travels, and the deduction to be made on account of centrifugal force.

Tensile strength of belting.—Leather belting of high quality, single thickness, of about one-fifth of an inch, will break upon the application of a stress of about

1,200 pounds per inch in width. Special material will reach 1600 pounds, while material of commercial quality fails to exceed 560 pounds per inch in width. These figures are obtained from solid leather, without holes or joints of any kind. The weight is applied with deliberation, avoiding such special delay as would suffice to test the effect of time upon the results. But the strength of a joint of any kind is liable to fall below the strength of the solid belt. Joints may be made nearly as strong by paring, scarfing, and cementing; and in some cases a belt may be made with every joint of this kind, the last one being usually made when the

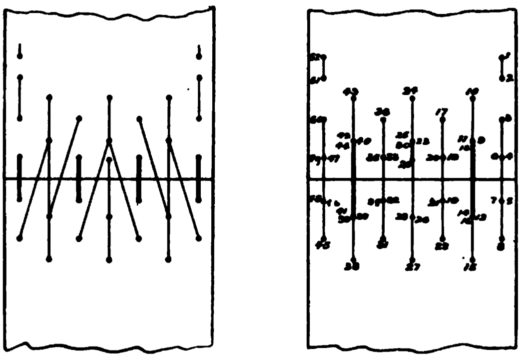


Fig. 107.—Joint of belt, arranged for strength.

belt is placed in position. In the majority of cases it is, however, found to be necessary to provide one joint in such a manner as to allow convenient adjustment in case the belt stretches; such a joint is always a weak part of a belt. Laced joints are the best for this purpose. The strength of an ordinary laced joint may be taken as 40 per cent. of that of the solid belt; but one carefully made and proportioned as in Fig. 107, may be loaded to 60 per cent. of the strength of the solid belt.

Assuming the strength of the belt to be 1000 pounds per inch in width, and taking a mean co-efficient of strength of joint, the nett strength of the belt is $1000 \times \cdot 50 = 500$ pounds per inch in width. The factor of safety for leather belting may be taken at 5. This gives the nett working strength of good single leather belting as 100 pounds per inch in width. Double belting is formed of two thicknesses of leather; in proportion to the sectional area it should bear twice the tension borne by single belting. In practice it is, however, found that the increase in strength cannot be relied upon to exceed about 80 per cent. The maximum tension in work which may be applied to good leather belts of double thickness is thus 180 pounds per inch in width, to cover primary tension, secondary tension, and tension due to centrifugal force.

Proportion of tension upon the two sides of a belt.—The proportion of tension to be applied to the slack side of a belt depends upon the frictional adhesion upon the pulley. When the belt and pulley surface are in good condition, the co-efficient of friction will not fall below ·22. Under these conditions, a belt which embraces one-half of each pulley will require a tension upon the slack side equal to one-half that upon the tight side. If the co-efficient of friction should reach ·35, the tension on the slack side may be allowed to reach one-third of that upon the tight side. In experiments made by Messrs. Briggs and Towne, the co-efficient was in every case found to reach or exceed ·3. But M. Leloutre, in perhaps the most extensive series of experiments made, found the co-efficient of friction to vary from ·09 to ·3. In practice it is, however, found that a belt will run well with the slack side under a tension of one-half that of the tight side, except when quite new, thus

showing that a co-efficient of friction of ·22 is perfectly safe, when everything is in order.

Working velocities of belting.—In ordinary practice main belts are driven at velocities of from 2000 to 6000 feet per minute. Subject only to correction on account of the action of centrifugal force, a belt will transmit power in proportion to the speed at which it is driven.

Tension upon belting due to centrifugal force.—Centrifugal force acts at all speeds in imposing a stress upon the belt, due only to its weight and the velocity at which it moves. This is imposed equally upon the slack side and the tight side. The amount of tension in pounds per inch in width due to centrifugal force equals—

$$\frac{\text{Weight of belt in pounds per square foot} \times (\text{velocity of belt in feet per second})^2}{12 \times 32 \cdot 2};$$

By subtracting this amount from the permissible stress upon the belt, the maximum working stress is obtained, which is to be treated precisely as though centrifugal force did not exist. As an example, a belt may be considered which works at a velocity of 4,200 feet per minute = 70 feet per second, the weight per square foot being 3 pounds. The total tension to be 180 pounds per inch, and the working tension on the tight side to be twice as great as that on the slack side. In this case the centrifugal force is 38·0 pounds per inch width, which, subtracted from 180 pounds, leaves 142·0 pounds as the sum of the primary and secondary tensions. In this case the sum of these is twice as great as the secondary tension, therefore the two are equal. The primary tension (71·0), multiplied by the velocity and divided by 33,000, gives 9·0 indicated horse-power transmitted per inch width of belt. The total tension to which the belt will be exposed in work will be 180

pounds per inch on the tight side, and 109 pounds on the slack side.

Centrifugal force has no *direct* action upon a belt when moving along a straight line, but only when moving in contact with a pulley. In the latter case it acts precisely as though it formed an endless belt, placed in loose contact with the entire circumference of a revolving pulley. The bursting tendency which arises is analogous to that in a revolving wheel. The centrifugal force due to any individual particle varies with the square of the velocity, and inversely as the radius. But an increase in the radius of a continuous ring is accompanied by an increase in the number of particles which give rise to centrifugal force. Consequently the effect of the radius is neutralized, and may be neglected.

Critical speed of belt dependent upon centrifugal force.—At low speeds the effect of centrifugal force is so small that it may be neglected, but at high speeds it becomes very important. At one speed, varying with the conditions, it reaches a point beyond which an increase of speed causes a decrease in the amount of power transmitted. For this reason it is generally unwise to run a belt at a higher speed than 4,500 feet per minute. In one case an increased diameter of main pulley may, however, be desirable to increase its effect as a fly-wheel. In another case the object may be to avoid the use of very small pulleys, on account of their prejudicial effect upon the belts. If it were possible to provide a belt of material which had absolutely no weight, the element of centrifugal force would entirely disappear, and any means by which the weight of belt is reduced give proportionate relief. Thus, in the example just given, a quality of leather might be adopted of strength to bear loading to 210 pounds per inch width, the weight per square foot remaining

the same. The superior strength of the belt allows a reduction in width, so that less weight is employed. The horse-power per inch in width rises from 9·0 to 10·9—an increase of 21 per cent., while the strength of the belt is only increased by 17 per cent. At higher speeds the difference is very much greater. An increase in strength of 40 per cent. can be easily secured from best makers, at a proportionately enhanced cost. Some makers also produce belting which, by reason of special processes and attention in manufacture, is lighter than ordinary belting, while being at least as strong. The use of such belting also effects an important reduction in centrifugal force.

Light belts are also less subject to oscillation arising from, and tending to aggravate, unsteadiness in working. They are usually less fitted for resisting the abrasion caused by contact with a belt-shifting fork, but this objection very seldom applies to main driving belts, for which purpose they are chiefly and with greater advantage applied.

Table of power normally transmitted by belting.—Table XXVII. is calculated upon a permissible total working tension of 180 pounds per inch in width. The first column gives the speed in feet per minute. The second and third columns refer to ordinary double belts weighing 3 pounds per square foot. The fourth and fifth columns refer to double belts of superior quality which, while amply safe to work at the given tension, only weigh 2½ pounds per square foot. The sixth and seventh columns refer to chain belting of equal strength, but weighing 6 pounds per square foot. In each section of the table is given the tension or stress due to centrifugal force arising from the given speed of belt and its weight; and in connection therewith the actual horse-power when allowance is made for centrifugal force. In

each case after the elimination of centrifugal force, the tension on the tight side is assumed to be twice as great as that upon the slack side. If the conditions applying to any particular instance are considered to be sufficiently

TABLE XXVII.—HORSE-POWER TRANSMITTED BY BELTING, PER INCH IN WIDTH.

Load, 180 pounds per inch. Arc of contact, 180 degrees.

Speed in feet per minute.	Good ordinary double belting, weighing 3 pounds per square foot.		Superior double belting, 2¼ pounds per square foot.		Chain belting, 1 inch thick, 6 pounds per square foot.	
	C.F. lbs.	H.P.	C.F. lbs.	H.P.	C.F. lbs.	H.P.
1,500	4·8	3·98	3·6	4·01	9·7	3·89
1,800	7·0	4·72	5·2	4·77	14·0	4·53
2,100	9·5	5·42	7·1	5·50	19·0	5·12
2,400	12·4	6·09	9·3	6·21	24·9	5·64
2,700	15·8	6·72	11·8	6·88	31·6	6·07
3,000	19·4	7·30	14·5	7·52	38·8	6·42
3,300	23·5	7·82	17·6	8·12	47·0	6·65
3,600	28·0	8·29	21·0	8·67	55·9	6·77
3,900	32·8	8·70	24·6	9·18	65·6	6·76
4,200	38·0	9·04	28·5	9·64	76·1	6·61
4,500	43·7	9·29	32·8	10·04	87·4	6·31
4,800	49·7	9·48	37·3	10·38	99·4	5·86
5,100	56·1	9·58	42·1	10·66	112·2	5·24
5,400	62·9	9·58	47·2	10·86	125·8	4·43
5,700	70·1	9·49	52·6	11·00	140·1	3·45
6,000	77·6	9·31	58·2	11·07	155·3	2·24
Lines in Fig. 108.		*A.*		*B.*		*C.*

favourable to ensure that the tension upon the slack side shall not exceed one-third of that on the tight side, the horse-power will be increased by one-third. If at any future time means should be discovered whereby

the ratio may become a fourfold one, the horse-power will be increased by one-half above that shown in table. If a new belt is proportioned in accordance with Table XXVII. and is at once exposed to full work, it may be necessary, as a temporary measure, to apply a secondary tension equal to double the amount of the primary tension, in order to prevent slipping. This should, however, not be allowed to continue longer than necessary. In the example previously quoted, the primary tension would be 71 pounds, the secondary tension 142 pounds, and the tension due to centrifugal force 38 pounds, equal to a total tension of 251 pounds per inch.

Reserve power.—Table XXVII. gives results which can be secured without difficulty in good average practice. Many engineers, in deciding upon the proportions of belts, adopt very much lower tensions, and thereby secure a corresponding increase in their useful life. When this can be done without difficulty, the step is to be commended. But difficulties often prevent this; the capital outlay is greater, and space is sometimes unavailable. A large number of belts in work are, however, loaded to a much greater extent than they are supposed, or than they require, to be.

Results shown graphically.—The horse-powers given in Table XXVII. are translated into graphic form in Fig. 108, in which the several lines refer to the following—

A. Good ordinary double-leather belting, weighing 3 pounds per square foot.

B. Double-leather belting of superior quality, weighing $2\frac{1}{4}$ pounds per square foot.

C. Chain belting, weighing 6 pounds per square foot.

D. An ideal case for comparison in which the effect of centrifugal force may be disregarded, owing to small weight of belt.

In each case a total tension of 180 pounds per inch in width is assumed to be applied to the belt.

Direction of belts.—All belt transmission arrangements work most efficiently when the belts run in a nearly horizontal direction, when the two pulleys are of nearly equal diameters, and when they are at a moderate distance apart, usually about 30 feet.

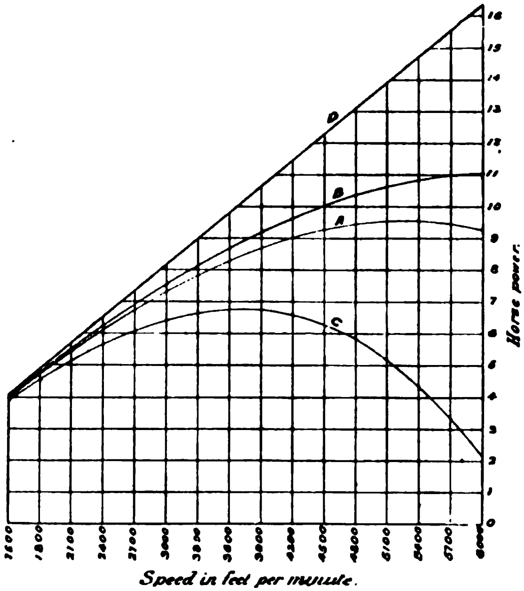


Fig. 108.—Power of belting, shown graphically.

The tension upon belts which approach the vertical direction, and which are of great length, varies greatly in different parts. In a double belt running over pulleys whose centres are 100 feet apart, the tension at the top is from 18 to 28 pounds per inch greater than at the bottom. In calculations, such figures should be treated exactly in the same manner as centrifugal force.

Applying this to the case before taken, it will be found that a belt of good double leather, 100 feet over centres, subjected to a maximum tension of 180 pounds per inch, and travelling at 4,200 feet per minute, will transmit, in a horizontal direction, 9·0 horse-power, but in a vertical direction only 7·4, showing a reduction of 18 per cent. The difference is greater when the belt is loaded, as in an elevator belt for raising material. Such a belt is driven with much greater efficiency by the upper pulley than by the lower one, on account of the better contact secured by reason of the weight. In the case of a long belt working vertically, the excessive weight imposed upon the upper pulley is transferred to the upper bearings, to the relief of the lower ones. In a horizontal belt the weight of the belt acts equally at each end, and its effect is lost in the primary and secondary tensions.

TABLE XXVIII.—CORRECTIONS TO BE APPLIED TO HORSE-POWER FOR ARCS OF CONTACT LESS THAN 180 DEGREES.

Angles subtended by arc of contact of belt upon either pulley.	Ratio of tensions upon two sides of belt with constant co-efficient of friction.	Ratio of horse-power which may be transmitted by same belt.
180°	2·00	1·00
165°	1·89	·94
150°	1·79	·88
135°	1·69	·82
120°	1·60	·75
105°	1·51	·67
90°	1·42	·59
75°	1·34	·51
60°	1·26	·41

Angular extent of contact upon pulleys as affecting adhesion.—The frictional adhesion of belts has been hitherto treated in connection with pulleys of which one-half of the entire circumference was embraced by the belts. A belt will hold equally well upon pulleys of very different diameters, so long as the arc of contact is measured by equal angles, whether these be more or less than 180°, or half circumference. The hold of a belt is reduced upon a reduction of the arc of contact, but such reduction is not in strict proportion to the angle measuring each arc. Table XXVIII. gives the significant details referring to various angles. The third column gives the proportions of power which may be transmitted when a belt embraces less than one-half of the circumference of either pulley. As an instance it may be noted that when the arc of contact is only 120°, the driving power of the belt is only 75 per cent. of that which it would possess if the angle were 180°. In the case of crossed belts the angle on each pulley must of necessity exceed 180°, but these are never used for main driving. The power is affected to precisely the same extent whether the deficiency of contact is upon the driving or the driven pulley. Numerous experiments agree in showing that the frictional adhesion of belting upon pulleys is independent of the diameter, but is reduced upon a reduction of the arc of contact. All these experiments have been conducted at extremely low speeds, with the weights steadily applied. Accurate experiments upon belts under working conditions would be most valuable, but would be difficult and costly to conduct.[1]

A strong impression prevails with practical men to the effect that adhesion upon a pulley depends upon

[1] Since this was written, the *Société Industrielle du Norde de France* have announced a series of trials of this kind to be made upon ropes and belts.

mere length of contact, irrespective of whether it takes the form of a comparatively small arc upon a large pulley, or a large arc upon a small pulley. As a general principle this is clearly shown to be erroneous; but it is impossible to declare that there is absolutely no ground for such impression. At a given speed, the length of time during which the belt remains in contact varies with the lineal measure of the arc of contact, and possibly this fact causes some increase in the holding power upon a large pulley. The wear and tear upon belts is greater with small pulleys than large ones. The adoption of liberal proportions based upon such impression is beneficial in affording relief to belts upon pulleys of small diameters. Consequently, though the impression fails to receive the support of direct evidence, it is entitled to some consideration. Evan Leigh's rule is based upon this impression, and has been adopted with marked success where good double leather belts are used, upon pulleys of 6 or 7 feet in diameter. It is based upon a primary tension of one half-pound per inch in width and per inch in length of contact. Applying this to the case in which two pulleys of 10 feet in diameter are connected by a belt travelling 4,500 feet per minute, the working tension on the tight side being twice as great as that upon the slack side, and the belt weighing 3 pounds per square foot, it will be found that the total tension upon the belt is 232 pounds, which is too great for belts of good ordinary quality. At low diameters the proportions err in the opposite direction, but give relief in the manner before described. It is therefore better to use the rule described in connection with Table XXVII., and make such allowance as appears necessary for small pulleys, or for unfavourable conditions which may arise. Obviously a pulley of large diameter will drive more than one of a

small diameter, working at the same speed of revolution, but this is allowed for by calculating upon the lineal speed of belt.

Thick belts objectionable upon pulleys of small diameters.—Belts of double thickness are occasionally required to work round pulleys of very small diameter, as in driving centrifugal pumps, circular saws, etc. Except in such cases of urgent necessity, double leather belts should never be used on pulleys of less diameter than 3 feet, and better if not less than 4 feet. Belts of single thickness are decidedly preferable where they can be employed, as they are more flexible, more durable, and absorb less power in working than in the case of double belts.

Distance of shafts.—When a belt is used to connect shafts at a greater horizontal distance apart than 60 feet, and the driving is at all irregular, it may be necessary to use guide-pulleys to steady the belt. This remark applies equally though the shafts may be at widely different levels. When the shafts are less than 20 feet apart, measured from centre to centre, the belts are liable to give trouble, and should receive special attention. While they are in good order they are subject to the same rules as though the shafts were further apart. But they are deficient in elasticity, and when they require tightening, the operation should be conducted with especial care to avoid cutting out too much at a time.

Accidental limitations of speed.—When arranging the general designs for a new installation, and in many other cases, there is a large choice of diameters of pulleys, and widths and speeds of belting, so that there is a probability that the best proportions may be decided upon. But in some cases, especially in connection with alterations to existing works, it is much more

convenient to provide space for pulleys of small diameter and wide belts, than the opposite. In such a case the pressure upon the bearings must not be overlooked. The speed of each line shaft is usually approximately determined with reference to the pulleys which drive the separate machines. The speed of the main belts should, however, never be allowed to fall below 3000 to 4,500 feet per minute if it can be avoided.

Speed calculations.—The revolutions of the two pulleys connected by a belt are inversely in accordance with the virtual radii of the pulleys. When there is a considerable difference in the radii and great accuracy is required, the virtual radius of each pulley is measured to the centre of the thickness of the belt. The speed of the driven shaft will fall a little below the calculated speed, by reason of creeping of the belt. The amount of this variation depends upon the loading of the belts, and cannot be exactly calculated. In heavy driving an allowance of three per cent. for this purpose will, however, usually meet the case.

Departures from cylindrical form in pulleys.—Belts are always intended to run in the centre of the pulleys, but for various reasons they deviate slightly from this condition in work, first running towards one side and then the other. It is most undesirable that a belt should project beyond the pulley in work, and consequently the width of the pulley is greater than that of the belt. The pulley rim is also made convex or rounded, or crowned in cross section, so that its diameter is greater in the centre than towards each side, with the object of ensuring that the belt shall not leave the centre of the pulley to any serious extent. The manner in which this is secured may be best understood from a consideration of the action of a belt upon a conical pulley, as shown in Fig. 109. The conical pulley

shown is of smaller diameter at *a* than at *b*. The belt is stretched more along the edge towards *b* than along the opposite edge. It thus assumes a curved form, and its lateral stiffness causes it to deviate towards *b*. A simple experiment with a conical earthenware jug and a strip of paper, 2 inches wide by about 20 inches long, will best illustrate this action in a simple way. The same action also takes place if the plain taper be replaced by a curved taper. Two pulleys with curved taper, placed with their larger diameters together, are

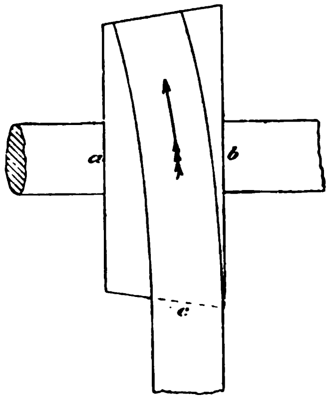


Fig. 109.—Climbing of belt upon a conical pulley.

equivalent to one pulley with a rounded face. If a belt which is running upon such a pulley be constrained by any cause to leave the central position, its equilibrium is disturbed, and only restored when the belt returns to its original position.

Departures from parallelism of shafts.—But the fact that a plain belt, running upon pulleys of variable diameter, which are mounted upon parallel shafts, will climb towards the largest diameter is often misinterpreted to show that when two shafts which lie in one

plane (say at one level), but are not parallel, are connected by a belt running upon pulleys, each of which is of uniform diameter throughout, such belt will tend to run towards that part of the shafts where the distance apart is the greatest; or in other words, will tend to tighten itself in running. As a matter of fact, however, an opposite action is developed. If the former assumption were correct the action of a belt-shifting fork would be a reversed one, so that a movement of the fork towards the right would cause the belt to move upon the pulley towards the left. But perhaps the most convincing proof (except that upon shafting actually at work) will be furnished by the same slip of paper as in the last experiment, applied to an object of uniform diameter.

Verification of accuracy in erection of pulleys.—Cases occur in which belts run upon shafts which are out of parallel in the manner described, but which are prevented from running off either by excessive crowning of the pulleys, or by other means sufficient to overpower such tendency, but which always fail to secure efficient working. In all belt-gearing work, accuracy of line, level, parallelism, and diameter of pulleys are of very great importance, and especially is this the case with belts of large size. When heavy main belts require the application of pressure upon their edges, in order to restrain them from running off, or when excessive slip takes place, some grave defect must exist in their arrangement. A line should be carried past both pulleys, just clear of the shafts. If the shafts are parallel, and the pulleys fair with each other, the line will just touch the rims in four places, and the same when tested on the opposite side. The shafts should then be levelled and tested for parallelism by direct measurement. The best test for diameters of pulleys

consists in the application of a steel tape or wire for measuring the circumference. This should be measured close to each edge of the pulley and the two compared. These means will be almost sure to show a sufficient reason for any difficulty which may have arisen in work.

Exceptional transmission by twisted belts.—Cases occur, especially with narrow belts, in which the shafts are not parallel, but in which the transmission is efficient. The simplest case of this is that of the "quarter twist" or "half-crossed" belts, largely used for driving separate machines by means of one shaft,

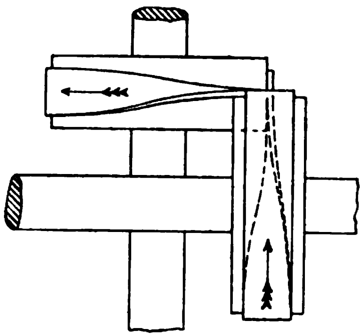


Fig. 110.—Plan of quarter-twisted belt and pulleys.

which is at a different level from the shafts of the machines, and whose horizontal direction is at right angles to that of the latter. In such cases the requisite condition to be observed is that the belt shall leave each pulley at a point which lies in the centre plane of the opposite pulley. Fig. 110 shows the arrangement of such pulleys in ground-plan. The two pulleys may be drawn upon a piece of paper, as in Fig. 111, in which the pulleys *a* and *b* are shown as though they were revolving in one plane, connected by an ordinary open belt, *c*, *d*, *e*, *f*. The line *g h* is tangent to each pulley,

of which one is upon each side of it. The bottom pulley revolves towards $g\,h$ on the under side, and the top pulley on the upper side. The paper may be doubled along the line $g\,h$, and the two parts placed at any angle with each other, until, when one pulley has turned half round, the two pulleys again become parallel to each other, and the case becomes that of an

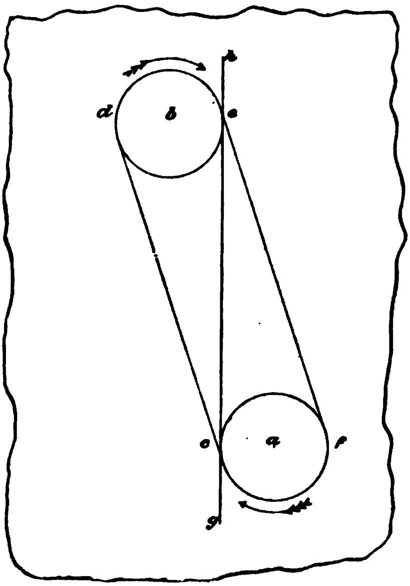


Fig. 111.—Diagram of twisted belt.

ordinary crossed belt. In the two extremes, or in any intermediate position, the two circles represent in position and direction the centre planes of pulleys upon which a belt will run quite efficiently, but only in one direction. If the direction of motion be reversed, the belt will at once leave both pulleys, except in the extreme cases in which the pulleys are parallel to each

other. It may be noticed that if each pulley had a groove turned along the centre of its face, a fine wire or string, corresponding to the line gh, would lie with almost absolute accuracy in such grooves when the pulleys are adjusted in position. The principle is best adapted for application to narrow belts. The maximum width in any particular case depends upon the diameters of pulleys and the distance apart; but the exact proportions admissible do not admit of definition by rule. In all such cases the wear and tear of belts is greater than that of ordinary open belts, chiefly owing to the fact that one edge of the belt is always under greater tension than the other. This difference in tension exists in a small degree throughout the belt, but is most strongly developed at the point where the belt leaves each pulley. For this purpose Mr. Gehrckens uses a compound belt made of sections attached to each other by overlapping. The pulleys for quarter-twisted belts may be rounded obliquely with advantage, the large part being in the direction towards which the belt is diverted from the plane of the pulley. Chain belting of unequal thickness may be used for quarter-twist work; the pins, being further from the pulley at the edge exposed to greatest tension, will tend to relieve such tension, but such a measure should be cautiously resorted to, or the tension may be relieved too far. Pulleys for quarter-twisted belts should be arranged so that they may be slightly adjusted by movement along the shafts, if in work they appear to require it by reason of peculiarity in belts or speeds. The power of quarter-twisted belts may be calculated in the same way as that of ordinary open belts, but allowing only from one-half to two-thirds of the usual tension, owing to the irregularity of its application.

Large heavy driving-belts cannot be worked as

quarter-twist belts, but a slight angle in the same direction may be sometimes introduced with advantage. In considering the possibility and utility of this, the loose paper diagram will be found to be of service, or it may be replaced by a pair of boards connected by hinges. Generally speaking, however, it will be found that rope gearing provides more convenient means for overcoming difficulties arising from small irregular angles. Rope pulleys should, if possible, be arranged in the same way as belt pulleys, but the conditions need not be observed with the same degree of accuracy.

Use of guide-pulleys.—It sometimes happens that a belt-gearing arrangement is required under circumstances which forbid the fulfilment of the conditions embodied in the loose paper diagram. In such cases the use of one, two, or more guide-pulleys becomes a necessity. Great latitude is permissible in the use of guide-pulleys, but it is necessary that in all cases the belt shall leave each pulley at a point in the centre plane of the succeeding pulley, and only from the most extreme necessity should guide-pulleys be adopted of diameter less than 12 inches, but greater if possible.

Very heavy driving belts should not be carried round a large portion of the circumference of small guide-pulleys, on account of the resistance thereby encountered. On the other hand, the contact should be sufficient to maintain motion, as if the pulley is allowed to remain stationary for a short time it is very likely to wear into flats and give continued trouble.

Importance of an efficiently balanced condition in guide-pulleys.—Small pulleys run at a very high speed of revolution, and if they are defective in balance, vibration will be caused, in proportion to the ratio which the error bears to the whole weight of the pulley and the shaft in proximity thereto. A guide-

pulley for belting is often very wide; it may be perfectly balanced when standing, and yet prove exceedingly unsatisfactory in work, owing to the fact that an error in excess at one end may be opposed by a similar one at the opposite end of the pulley, and on the opposite side of the shaft. When the pulley is in rapid revolution, the combined action of these errors is to impose a violent wrench upon the shaft, alternating with each half revolution. For this reason, all details intended to be worked at a high speed should be rough turned all over, or equivalent means resorted to for securing exact symmetry, at least so far as weight is concerned. Every detail should be balanced within itself, and when unavoidable departures are necessary, the weight of such should be balanced as nearly as possible in the same plane of revolution; this should be done before the addition of any other detail, large or small, upon the same shaft. In high-speed fittings of all kinds, labour expended in the attainment of an efficiently-balanced condition is exceedingly well invested. Even the largest pulleys will work unsteadily if slightly defective in balance. This question is further treated in a separate chapter.

Textile or fibre belting.—Belting of leather has been, and still is, used more extensively than any other. But belting is also very largely used made from various vegetable and animal fibres, of which perhaps cotton and hair are the most important. India-rubber belting is used for special purposes. The strength of best cotton belting, of thickness equal to that of single leather, is about 800 pounds per inch in width. Hair belting is of about equal strength. These are, however, lighter than leather of good ordinary quality by about 20 to 30 per cent., according to the amount of composition with which they are treated. Cotton belts are

often made by doubling together and sewing in several thicknesses one wide piece of specially woven canvas, with composition interposed between the folds; they are hence described as to strength and thickness by the number of plies. In other cases, belts are made in a manner strictly equivalent to this, but the connection is made by yarns, controlled by the loom which weaves the mass of the belt. In other cases, again, the whole mass is woven together, so that no distinction between the plies is possible, but the belt is still described by the number of plies. An obvious objection to this is that different makers may supply a very different belt under the same designation. A much better standard would be the weight of pure fibre per square foot and its description. Hair belts are generally woven similarly to cotton, in one wide solid web. Hair yarn is, however, coarser than cotton yarn. Cotton and hair belts will not withstand the same amount of mechanical rough usage as leather. They are consequently not so well adapted for use in connection with fast and loose pulleys, on account of the chafing of the edges which is caused by contact with the belt-shifting fork. Improvements are, however, made from time to time which reduce the disadvantages under which they suffer. Messrs. Tullis insert a strip of leather in each edge of a belt, to take the wear of the fork. The Rossendale Belting Company, in addition to the general treatment of all ordinary belts with preservative compound, use a special compound for the edges of such as are exposed to chafe by strap-fork or otherwise. Fibre belts are much better adapted than leather ones for exposure to the weather, or to moisture derived from material with which they come into contact, or steam. Hair belts are especially well adapted for use in hot, damp climates. When saturated with a waxy compound, they are used

for conveying wet coal or grain from washing machines to great distances. Under such work, leather would deteriorate very rapidly. A further advantage is the great length of fibre belt which can be obtained quite uniform and free from joints. The very moderate cost of cotton and hair belting is usually an advantage. But in an accurately proportioned piece of work the belts will be rather wider than if of leather; consequently the pulleys are more costly. The total cost will, however, depend very much upon the special conditions of each case. As a general rule it will, however, be found better to adopt fibre belts rather than leather ones of doubtful quality; but that leather ones of high quality will be better than either, and less costly when considered over a series of years. All belts tend to slip when new, and suddenly exposed to full work. In this respect cotton belts are worse than leather, but after a short period in work, a belt of any kind becomes covered by a film of deposited dust, etc., when very little difference in tendency to slip exists between different belts exposed to the same tension.

Examination of textile belts.—Belts of cotton and hair may be examined by dissection. In doing this the threads which run longitudinally should receive most attention, as they have to perform the heavy work. The cross-threads are of importance in belts which are likely to be subjected to chafe. In either case the fibres may be separated by a little care, and a comparison made between those of different samples, as to length and strength of ultimate fibre. In comparing cotton with hair the individual fibres in the latter case will be found to be both longer and stronger than in the former; but the hair fibres are thicker than those of cotton, and consequently a less number is used.

Preliminary stretching and subsequent shrinkage of

belts.—Belts of all kinds, before being gradually put to work upon new shafts, should be exposed to tension for several days. The amount of this tension should exceed that to which they will be exposed in work by from 30 to 50 per cent. This will cause an increase in length, but on the removal of such tension the belts will tend to slowly resume their original length, so that no time should be unnecessarily lost in jointing them in position upon the pulleys. This shrinkage is largely promoted by the presence of damp, so that if at any time a belt is run off the pulleys and left off for some time, it should be kept dry. The additional tension previously referred to as necessary for new belts applied to *full work*, will effectually perform the necessary stretching in many cases. The journals, if in good, smooth, working condition, will sustain the additional pressure, provided that special attention be paid to the lubrication for the time. In a cotton belt the stretching and shrinkage take place in a straight line, but this is not always the case in a leather belt. Good makers are, however, exceedingly careful in selecting the leather for any belt, so as to give the greatest possible degree of uniformity both in strength and thickness from end to end, and also from one edge to the other. When this cannot be directly and sufficiently secured, the leather is pared to one uniform thickness, so as to give uniform elasticity; otherwise the parts thinner and weaker than the rest become more stretched; the working strength of the belt suffers; the variation in thickness is magnified; and the belt works with great irregularity. Uniformity of weight is also of importance, as affecting steadiness of working.

India-rubber composition belts.—Belts made of several plies of canvas, cemented together by india-rubber composition, and faced on one side and over the edges

with the same, are used in wet places and for conveyance of grain. The india-rubber composition is ruined by contact with oil of any kind. It will also crack and break off if of unsuitable composition, or if applied without proper attention to the condition of the canvas. Belting of this kind is very flexible, working easily round small pulleys or rollers, which is a point of prime necessity in connection with its application to granary work. It is very elastic, and possesses high adhesive power for driving. For grain-conveying belts this type has proved most successful, but it is rapidly becoming displaced by cotton belting, which answers nearly as well and is much less costly. Cotton belts for this purpose are faced with a film of gutta-percha or other gum, deposited from solution, and applied with a view to improvement in adhesion and cleanliness of surface.

Advantage secured by the use of belts of high quality. Belts of all kinds are in every way inefficient and really costly in use, unless the material, disposition, and workmanship receive great attention. As a rule, though not an invariable one, the price gives a good indication of the value, and the best is really the cheapest. Belts which are marked at intervals with the maker's name are generally to be depended upon, and the date might often also be added with advantage.

Jointing of leather belts by splice.—Belts of leather must be made from pieces of such a size as can be obtained from a hide; these are invariably connected together by means of spliced joints pared to a feather edge. The entire length of each joint varies according to the width and thickness of each belt. The joints are closed with cement, and if this is properly done the work is quite secure. The two parts of each splice should be well matched in substance, so that they will

stretch equally, and the paring of the joint should be accurately performed. After cementing, the edges should be examined, and if any slight ridges are found they should be pared over so as to remove any tendency to start a tear in the cement joint. Splices which are unskilfully made are very likely to tear apart in the cement, if not secured by additional means. These may take the form of sewing with laces, wax thread, or wire, or riveting with round, flat-headed, copper rivets and washers, or with plain wire clench-pins, or thin metal plates riveted over the edges of the joints. All these involve a reduction of strength in the belt, by reason of holes cut through; and they are quite unnecessary if the splices are properly made.

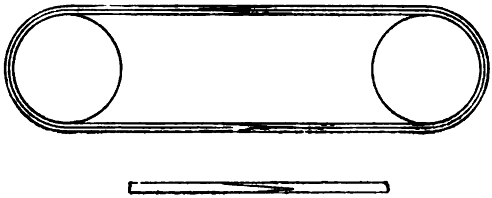


Fig. 112.—Direction of belting splice, with respect to motion.

Direction of splices.—In the best double belts care is exercised to ensure that the points of all spliced joints are directed towards the same end on both faces throughout, as in Fig. 112; then the end towards which the joints are directed is arranged to follow the other, when the belt is ultimately set to work. Obviously, with plain spliced joints in a single belt, this cannot be done on both sides, but Messrs. Tullis secure the same object by means of a joint of bird's-mouth form, shown to a larger scale in Fig. 112. In the absence of some such arrangement, the joints are set to run the proper way on the inner face, this being the most important one,

especially for crossed belts. If a guide-pulley is applied —as is often the case—to the outer side of such a belt, the corners of the joint on that side usually begin to rise sooner or later, when it is almost impossible to re-secure them. With narrow belts the best way is to twist the belt through one half revolution, or turn it over opposite to any guide-pulley which would otherwise press against the back of the belt. The belt will thus run with its face side against all pulleys with which it may come into contact.

Connection of layers in double belts.—A leather belt of double thickness may have the two portions attached to each other by cementing only, by sewing, or by any kind of riveting; the first is, however, quite sufficient if properly performed. A large number of excellent belts have been made by cutting the leather into narrow strips and arranging these in the belt to break joint both longitudinally and transversely. These are excellent belts but costly, and it is found that by the adoption of care in selecting and joining the portions of a belt in the usual manner, all serious trouble is avoided. Occasionally a belt is used of single thickness over the main part, but doubled by a narrow strip along the back at each edge. Belts of cotton, hair, and india-rubber are manufactured in one continuous length, so that only the closing joint is required.

Chain or link belts of leather.—Leather chain belts are largely used for special purposes. In these the whole belt is composed of small links or pieces of leather jointed together by pins of wire, running across the belt and riveted at each side. The general arrangement of such a belt and of each link are shown in Fig. 113. Each link comes into contact with the pulley on its edge. Its ends are rounded just sufficiently to give the required degree of flexibility to the belt, but

leaving sufficient surface to bear upon the pulleys in work. In all cases, the pins should maintain a straight line when the belt is in close contact all across the pulleys. If the belt works upon pulleys with cross rounding, the distance from the face of the pulley to the centre of the links must be varied to suit the rounding. It is obviously necessary that all pulleys which come into contact with the face of any particular belt shall be exactly alike in cross-section. In ordinary cases, the back and front of such a belt are made alike, so that the belt may be turned over when partly worn; but if this is done it is better to do it at a comparatively

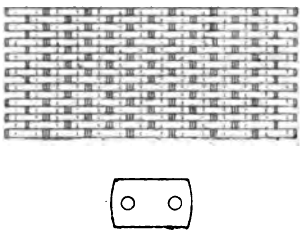


Fig. 113.—Linked belting.

early stage and repeat it, rather than allow the belt to wear half through its life before making any change.

Chain belts are jointed upon the pulleys after adjustment for length, by bringing the two ends together, then passing a skewer through the holes, following this with a rivet and clenching on each side. Such belts are therefore absolutely uniform in weight, strength, appearance, flexibility, and every other quality.

Weight of chain belting.—Under many conditions, chain belting gives a most efficient drive, but the fact that its weight is about twice as great as that of ordinary belting, becomes a decided disadvantage in the case of very long belts, especially if they approach the vertical

position. It is, however, an advantage in short horizontal belts. In all cases it is disqualified at a much lower speed than plain belting, by reason of centrifugal force due to weight, as already shown in connection with the power of belts. The cost of chain belting in proportion to weight is much below that of

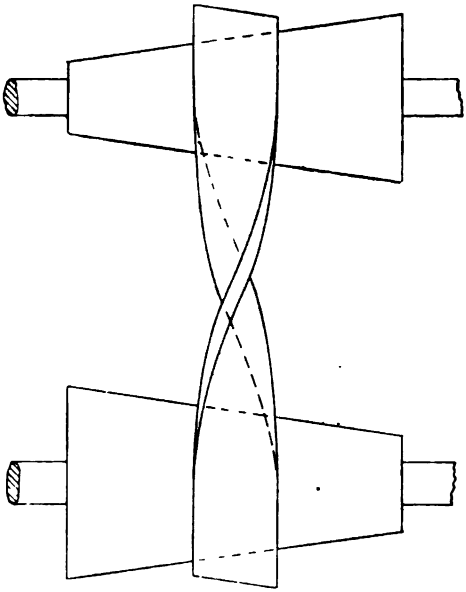


Fig. 114.—Taper cone pulleys, connected by linked belting.

plain belting, owing to the moderate cost of its manufacture and the completeness with which the material is utilized.

Lateral stiffness of chain belting.—The lateral stiffness of chain belting is much less than that of solid belting, consequently when it is adopted the pulleys must be erected with fair accuracy.

Special application of chain belting.—A very efficient

application of chain belting has been made by Messrs. Tullis, for the purpose of connecting a pair of taper cone pulleys, as in Fig. 114. This is most severe work for plain belting, but chain belting is made of taper sectional form, and the belt subjected to a half twist in each part, so that the pins upon each cone are parallel to the shaft, and all slip and irregular strain avoided.

Chain belting is not well adapted for use with strap-shifting forks, but protecting wings may be added for the purpose. These, with wide and well-rounded surfaces on the forks where they come into contact with the belt, and with reasonable use, will work fairly well, but should only be adopted for special reasons.

Closing joints of belting.—Belts are almost always supplied by the maker in one continuous length, ready for placing in position and completing by securing the two free ends together, so as to form an endless belt of the required length. Such a joint is usually the weakest part of the belt, and hence the importance of bestowing a due consideration upon the question, especially when the belt is intended to be heavily loaded. In comparing joints of various types, regard should be paid to the following conditions—

1. Strength.
2. Flexibility.
3. Durability.
4. Uniformity of weight and substance.
5. Uniformity of surface beneath belt.
6. Uniformity of surface above belt.

As compared with the main body of the belt, including the permanent joints.

7. Necessity for skill in process of jointing.
8. Facility for taking apart to remove.
9. Facility for taking up stretch.

Closing by scarf joints.—In a single or double leather belt, the closing joint may be made by paring, scarfing, and cementing. If a skilled workman is available for this purpose, the joint will be as strong and good in every respect as the rest of the belt for its work. This will be the case without any reinforcement by sewing, riveting, etc. It is very deficient as to points 8 and 9, but the degree of importance to be attached to these varies with each case.

Precautions to be observed in the insertion of holes in belting.—Joints of all other types require the insertion of holes in the belt, for the admission of laces, wire or twine sewing, bolts, rivets or other fastenings. In any case these holes must involve some loss of strength, according to the care exercised in the operation. The holes should be sufficiently large and numerous to admit fastenings, of which the strength is approximately equal to that of the substance of the belt which remains after the completion of the joint. If the holes are larger, or more numerous than to fulfil this condition, the belt will be weakened in its most vulnerable part. A hole for a lace should be so small that the lace can only be just drawn through, and similarly with bolts. In leather, small holes should be made circular, as also those required to fit bolts or rivets, as these are usually made of a circular form. Other holes should be made oval, with the length in the direction of the length of the belt. They are often cut with a sharp chisel or similar flat cutting tool, crosswise in the belt; this permits a thin lace to lie snug, and gives a good appearance, but involves a serious sacrifice in strength. Holes of this class are usually made for the laces, which are used to reinforce the cement in long splices, and are not very close together in the width of the belt, otherwise the belt would be ruined by them. In

the formation of the necessary holes in a leather belt, for the purpose of making a strong joint, some material is removed. But in woven belts of all kinds, a pointed tool may be inserted in such a manner as to make a hole without severing any of the threads of which the belt is composed. The tool may be either plain or combined with a smooth tapered screw point for convenience in use. If any threads are cut, the strength of the belt suffers to a corresponding extent.

The disposition of the holes made in a belt is also a matter of importance. The principles of strength which affect the disposition of the holes in riveted work of iron or steel, also apply strictly to this work. A certain number of holes is necessary to avoid loading each hole too heavily. The strength of the belt can only be maintained by distributing the holes so that a comparatively small number are placed in the line which is farthest from the end of the belt.

Simple overlap joint.—The simplest joint of great strength is one in which the belt is cut about 2 feet longer than is necessary to allow the ends to meet. The two ends are overlapped in proper relation to each other, to give the correct length of belt, and secured together by strong leather laces, copper rivets, screw fasteners, or otherwise. By this means a joint may be made for working over large pulleys whose strength and powers of durability are practically equal to those of the entire belt. But when working over small pulleys such a joint is liable to suffer from the repeated bending action to which it is exposed in work. The double thickness of belt, even when relieved by a partial paring, causes a difference in the velocity ratio, and a consequent jerk each time the joint reaches or leaves each pulley. The magnitude of such a jerk depends upon the speed of the belt, the character of the

work done, and, perhaps mostly, upon the diameter of the pulley. The excessive weight of such a joint also largely increases the irregularity of motion of the belt. Such a joint, in a belt driving a large centrifugal pump or a circular saw, is most unsatisfactory. Sometimes a belt is cut off to butt, and a loose apron piece placed over the joint, one-half over each part, the whole being sewn together as before. This is one remove better than the last, but still very unsatisfactory for important work.

Butt joint with iron cover-plate, bolted.—A useful joint to be applied to heavy belts for some purposes, is made by an iron plate on the back of the belt, bent nearly to the curve of the pulleys, and secured to the belt by fang-bolts. Owing to the necessity for flexibility, it is impossible to arrange the bolts in such a joint otherwise than in one line for each part, which leads to a sacrifice of strength and durability. The weight and stiffness of such a joint causes a slight knock when it reaches each pulley, but upon pulleys of large diameter it does not cause any perceptible check in the regularity of turning. The same kind of plate, when used only with leather belting, is occasionally secured by screws, after the manner of wood screws, but this is an inferior and weaker arrangement. It appears to cause less loss of tensile strength in the belt, but a calculation will show that, as ordinarily arranged, the screws themselves constitute the weakest part of the joint.

Butt-joint divided cover-plates, bolted.—Jackson's fastener consists of a malleable casting, secured to each part of the belt by a separate fang-bolt. A sufficient number is used to correspond with the width of the belt. Care should be taken that the total area of metal in the bolts is sufficient to bear the required tensile stress to be imposed upon it in screwing up, and the

shearing stress due to the pull of the belt. The holes must also be made with such accuracy as to give each bolt its fair share of the work.

Butt-joint with iron cover-plates, secured by prongs.—Harris's fastener consists of a plate of malleable cast-iron, with prongs cast upon the under side, upon which the belt is driven by means of a hammer. Another fastener, attached by prongs in the same way, is divided into two parts, which are attached together by hooking. This gives facility for taking asunder, and also imparts flexibility. The hinging centre should lie below the surface of the belt, or the parts will probably be tilted out of their hold. In both cases the fasteners are unsuited for use with belts whose texture is not sufficiently dense to give a good hold upon the prongs, and they are also unsuited for wide belts.

Raised or flanged joint, bolted.—In many cases the two ends of the belt are turned up at right angles to the general surface, then brought into contact, an intermediate thickness being interposed, and the whole secured together by bolts or otherwise. Means are required in addition to the bolts to maintain the belt in its form. Of this class, one variety is shown in Fig. 115, in which a pair of specially-shaped washers or plates, held together by bolts, is used. This fastening is repeated as often as necessary in the width of the belt. In this arrangement the tension upon the belt acts upon the bolts with an obvious amount of leverage. Taking the bolts as spaced at $2\frac{1}{2}$ inches centres across the belt, assuming that the leverage is in the ratio of 2 : 1, and that the bolts may not be loaded beyond 4 tons per square inch of nett sectional area, it will be found that $\frac{7}{16}$ bolts will barely suffice without allowance for initial stress in screwing up. Allowing for the latter element, the diameter of bolts of Whitworth pitch

should be $\frac{1}{2}$ or $\frac{9}{16}$ outside threads. The provision of separate washer-plates for each bolt allows sufficient transverse flexibility to meet the cross rounding of pulleys. There is no deficiency of longitudinal flexibility, and the tensile strength is fairly well maintained, provided that the belt is of a nature to stand bending round the roots of the washer-plates. The heavy projecting mass of the joint quite forbids the use of outside guide-pulleys, and slightly militates against

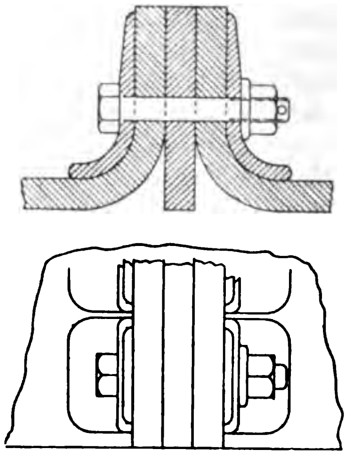


Fig. 115.—Flanged and bolted joint for belting, working at moderate speed.

steady working in any case. With pulleys of small diameter and high speed its use is quite inadmissible.

Link fasteners.—Small yellow metal fasteners are often used for small belts, and require the ends of the belt to be turned up in a manner similar to the last. These are very convenient in use and fairly strong, but are often supplied in such a rough condition as to cause very great chafing of the belt. Chafing is, however, often largely increased by the use of fasteners

of such a size as to allow rubbing. On account of the small size of these fasteners, there is no necessity for the removal of much material in making the holes in the belt, and when carefully applied they answer very well for light belts.

Jointing by laces or thongs of leather.—Belts of all sizes are often jointed by laces of leather. To effect this it is necessary to make a number of holes in each end of the belt; as already mentioned, these should be properly arranged for strength. Fig. 107 shows the two sides of a well-proportioned joint of this kind. All knots, ends, or slight irregularities should be kept on the outside, as also any parts of the laces which do not run strictly parallel to the length of the belt. This is chiefly with a view to mitigate the wearing effect upon the laces, which arises from the slight creep of the belt upon the pulleys. The oblique parts of the laces are divided so that half stand each way, thus avoiding the liability to draw the two parts of the belt out of line. Long unsupported loops are also avoided. For instance —one part passes from hole 3 to hole 5, but on the way is passed through hole 4. The numbers shown in the figure give the order in which the lace is passed. This is an excellent joint for use in difficult cases, upon any size of pulley, and at any speed. Several laces will be required for a wide belt, which may be connected by knots on the back surface. The outside of the belt may take a light bearing upon a guide-pulley. It may be observed that a belt generally passes on and off a guide-pulley at the same tension; there is therefore not the same tendency for the belt to creep upon the pulley, as occurs when work is transmitted to or from the pulley. Hence in such connection there is but little objection to oblique parts of lacing. A second form, more simple but less perfect than the above, is

shown in Fig. 116. A third arrangement, recommended by Messrs. Tullis, is shown in Fig. 117. This joint avoids any possibility of starting a crack through a number of holes by reason of neglect; it, however,

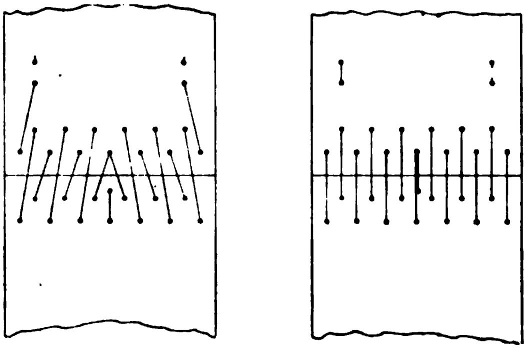


Fig. 116.—Laced joint for belting.

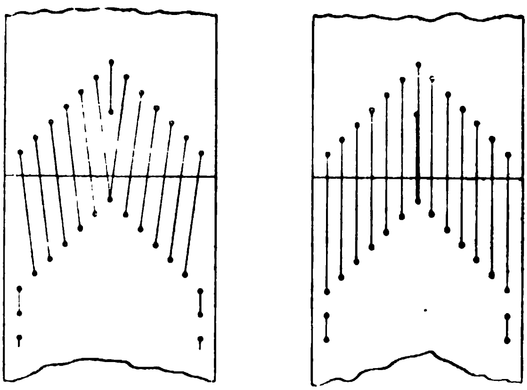


Fig. 117.—Laced joint for belting.

involves a great unsupported length of loops in the lacing. The firm just named very wisely advocate the use of laces in the jointing of all important belts.

Jointing of textile belts.—Most of the types of belt-

joints just described are equally applicable to leather, cotton, or hair. If, in the latter cases, there is any tendency to fray at the ends, a strong double sewing of waxed twine may be applied. Though the manner of jointing by laces is quite applicable to cotton belts, the texture of cotton prevents the laces from becoming so completely embedded below the surface as in the case of leather belts. Hence the laces in such joints wear rapidly and call for more careful watching and frequent renewal as a precaution against accident. It is obviously impossible to pare a woven belt to a uniform taper for splicing. But a belt made in plies which can be separated without serious loss of strength may have the several plies cut in step form, as in Fig. 118, and placed together so that the thickness of the joint at any

Fig. 118.—Stepped joint for sewing, applicable to textile belts made in plies.

part is only one ply greater than the main part. The two parts may then be carefully sewn together, giving a joint of strength and flexibility sufficient for almost any purpose. Cotton belts were formerly supplied with one end woven to taper form in the loom. Obviously such belts cannot be obtained from stock, and the practice appears to have been discontinued.

Cotton and hair belts have been lately jointed by means of an outer cover of double raw hide, treated with oil. This is carried across the belt about 3 inches in width, and secured by strong triple claw fasteners, which are clenched over in the manner of an ordinary paper fastener. When the cover and fasteners are well proportioned, the joint possesses considerable strength while new and tight, but it must receive regular attention.

Wide belts obtained by connection of narrower belts.—When a belt is required of greater width than can be conveniently obtained, two narrower belts may be employed, connected at intervals by Jackson's fasteners, already described. In this way no sacrifice in efficiency is entailed, but the two portions should be brought together with the cross-joints separated.

Jointing of india-rubber composition belts.—India-rubber belts are usually jointed by sewing with strong twine. In some cases they are cut square across; but more usually as in Fig. 119, or along a diagonal line, so

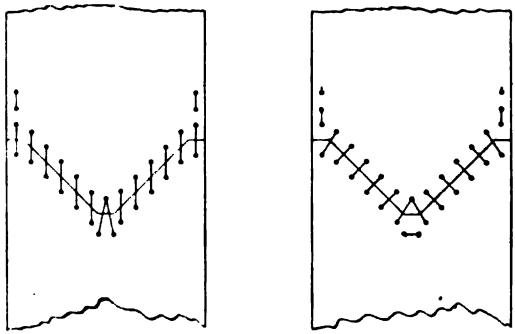


Fig. 119.—Sewn joint for india-rubber composition belt.

that only a small portion of the joint enters upon the pulleys at the same moment. In this case, also, the inside parts of twine should run nearly or quite parallel to the length of the belt. The outside of the belt should, however, be left as smooth as practicable when rolls are applied to it.

Precautions for safety during operation of jointing and replacing belts.—Light belts are always jointed when off the pulleys, and afterwards run on without stopping the engine, though it may be slowed down. In all operations of this kind means should be adopted to

secure the fullest measure of safety to the man and to the machinery. Belts are often cut and machinery strained, also personal injury inflicted, by assuming too much risk in such operations. Appliances for the furtherance of safety in this and many other respects are described in a work entitled, *A Collection of Appliances and Apparatus for the Prevention of Accidents in Factories,* published in 1889 at Mulhouse, price eight shillings.

Clips for use in jointing belts.—Large belts are always jointed when the pulleys are stopped. Each end of the belt is held in a pair of clips, the two pairs of clips

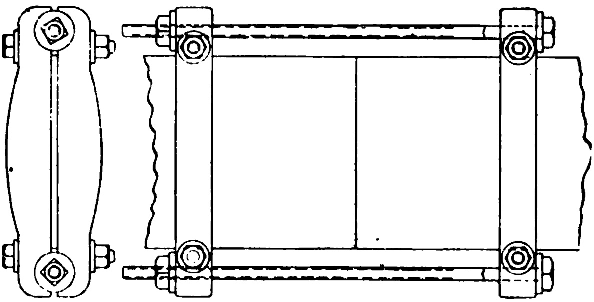


Fig. 120.—Clips for use in jointing belts.

being drawn together by screws, as in Fig. 120. The clamping bolts should be rather close to the belt, so as to give a good grip. The clips should be strong, and worked to such a face as to give tight contact throughout; this is best secured by trial. The drawing-screws are not heavily loaded. Square-threaded screws are best. But with either square or Whitworth threads a quick pitch is of great value in saving time. For this reason the screws may be made of larger diameter than would be necessary on account of mere strength. They should also be provided with long square shoulders, to

prevent turning round. In placing a new belt, or before cutting an old one to shorten it, the first operation is the marking of one end square across. The clips should be placed so as to allow convenient space for closing the joint after tightening. When the belt is sufficiently tightened for work, its accuracy of line should be tested, and if necessary, one or other of the screws adjusted to bring the belt to a perfectly straight line. The second end should then be cut, after marking, from the first end, which has been cut according to the original squared line. This gives a more correct result than by attempting to cut both ends by square.

Importance of cleanliness.—All belts should be kept clean, not spasmodically, but at all times. When more than a mere film of dirt is allowed to accumulate on belts, it tends to absorb the oil or composition, and to harden the substance of the belt, leading to inefficient contact with the pulleys, and liability to damage by cracking. Dirt may be removed by the application of warm water, if necessary adding a little soda. After allowing time for soaking, the dirt may be easily scraped off and the surface moisture removed. Then the belt should be allowed time to dry before working again.

Physical condition of belting.—Leather belts have various oils and greases worked into them during manufacture, which impart a soft, supple nature to them, which is a condition necessary to good contact upon the pulleys. In a new belt this is usually in slight excess, so that only after a short time in work does the belt attain good working condition; and many days may elapse before this is achieved with a belt which contains a large excess of oil. The condition of maximum efficiency continues for a period varying with the temperature and state of dryness of the air to which it is exposed, and with the severity of the work imposed

upon it. But after a time the loss of these substances causes the belt to fall off in condition, and to become hard. When this is the case, the loss should be at once made good by the application of castor oil, neatsfoot oil, boiled linseed oil, or cod oil and tallow. These should be applied to the outside of the belt, in quantities sufficient to restore the condition of the belt, but not to cause a film to rise on the working face, as such a film is likely to cause slipping in a heavily-loaded belt. When a belt is carefully maintained in most efficient condition, so as to give the maximum adhesion upon the pulleys, it may be worked in a less tight condition than would be otherwise necessary, which is directly conducive to a long life. The easy flexibility which usually accompanies such a condition must also secure some saving in the loss of power due to bending the belt round pulleys.

Cotton belts are soaked with a preservative compound largely composed of linseed oil, at the time of manufacture. In use they should be occasionally treated with boiled oil, in the way described for belts of leather.

The substances referred to will be usually found to be quite as efficacious as those supplied under special names for the purpose. The general adoption of barring-engines at the present time is a great convenience in respect to the cleaning and examination of belts and ropes.

The access of quantities of mineral oil to belts of all kinds leads to slipping, and physical deterioration of the belts, without securing any advantage. Such oil should therefore be removed at once by means of an application of powdered chalk, after which the belts should be cleaned. This trouble may arise from some accident to a bearing lubricated with such oil, or from

continued slow drip, and measures should be taken to prevent it. Oil or grease of any kind, and especially if of mineral origin, is highly and permanently injurious to all belts of india-rubber. All belts should be kept as free as possible from damp and from exposure to the weather. But leather belts suffer from these causes much more severely than cotton ones, which again are not so good as hair ones.

Periodical examination of belting.—All main belts should be subjected to frequent examination as to their general condition, the condition of joints and general freedom from chafe, or other cause for deterioration. Such examination should be always entrusted to one man, and the results entered in a book, as also notes upon any operations which may be undertaken as a result of such examination.

General conditions of application of belting.—By the adoption of belt gearing in place of a transmission by toothed gearing a great improvement is usually effected in absence of noise. In many cases the elasticity of belting is a most valuable feature in tending to reduce irregularities in driving. Usually, however, belts for heavy driving occupy much space, which in some cases can only be provided at great cost and with difficulty. In new work a very wide fly-wheel is usually provided to receive a number of separate belts side by side upon its face. Each belt is led to a separate shaft, which it drives by a pulley quite independently of the existence of other belts. In some cases, with a view to avoid the use of very long belts, the more distant shafts are driven from one or more of those less distant, the intermediate shafts being driven by belts proportioned to the combined amount of power. Leather belts of double thickness are little more than half the width of those necessary if made of single

thickness, and consequently the former are usually employed. Hitherto belts of treble thickness have not been often employed, but there is no reason why they should not be. Cotton belts are employed up to a thickness of half-an-inch. But though a direct increase in the thickness of belts has not been largely adopted, one belt may be run upon another without any attachment between the two. If guide-pulleys are applied to the outside of such a series, each belt must be of the same width, but otherwise the outer belt should be made slightly narrower than the under one. In like manner three or four belts may be superposed upon each other. The principle may be embodied in either one of two ways. Each belt may be treated independently, and replaced by two or more whose widths amount to little more than that of the original belt, the space occupied being correspondingly reduced. But if this would bring the belts to a very narrow width in proportion to their length, it should be avoided. In the second alternative system, several belts are placed upon each other and led to different shafts, the inner belt to the nearest shaft, and so on in order. Either of these measures may be adopted with cotton belts. In any case it is obvious that all belts except the outer ones must be so jointed as to present a surface free from any considerable projections. In either case the proportion existing between the tensions upon the slack side and the tight side of each belt remains practically unaffected, and consequently the ordinary rules for power are applicable, though some little amount of allowance is desirable.

In connection with all belt gearing a great pressure is imposed upon the bearings, by reason of the total pull of the belts upon the pulleys. This necessitates large surfaces in the bearings, and has hitherto caused

an important loss of power by friction. This loss is, however, very much reduced in good recent practice by improved means of lubrication, and therefore need not constitute any serious objection against the principle. The pull on the bearings is certainly greater than is the case with rope gearing. But in belt gearing there is no important loss of power, corresponding to that caused by the insertion into, and the withdrawal from, the grooves of the several ropes. Also several ropes working in conjunction rarely do so with the same degree of harmony as the several parts of one belt.

Belt gearing and rope gearing equally possess an important advantage in the possibility of an appreciable longitudinal motion of shaft, by reason of which the bearings are kept in a very much smoother condition than is possible in the absence of such freedom.

In some cases, especially in alterations undertaken with the object of a partial replacement of toothed gearing, a combination of toothed gearing and belt gearing is adopted. A second-motion shaft is generally driven by means of spur gearing, and the power distributed from it by means of belt gearing. In all such arrangements, the teeth of the wheels must be accurately made, to such form as to ensure absolutely uniform transmission. If this point is neglected, the vibration which arises appears to act most prejudicially upon the adhesion of the belts and cause slipping. To avoid this slip an excessive tension is imposed upon the belt, which may or may not fulfil the intention, but which is sure to lead to trouble in connection with the wear and tear of the belts. In such cases, almost of necessity, the belting speed is very much greater than that of the spur gearing, so that any defect which may be possessed by the latter is magnified in transmission, excessive backlash is sure to become developed, and teeth will be

broken out. It is therefore not surprising that an impression prevails that toothed and belt gearing are antagonistic in their natures. Where possible, it is indeed better to avoid a divided plan. This also fully applies to rope gearing in the same sense.

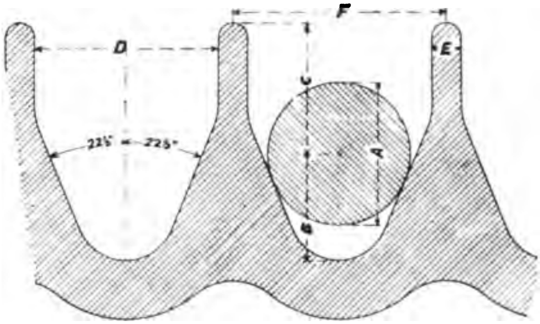
In reply to the question often put, as to whether toothed gearing, belt gearing, or rope gearing is the best, it is quite impossible to give a general answer. Each case can only be decided upon the fullest consideration of all conditions; when this is carefully done, it will be found that a field exists for all three systems. In all gearing installations weak points are liable to develop after the lapse of many years, and whenever repairs are undertaken an attempt is made to eliminate or obviate such weak points, with the result that, as a very general rule, well-conducted repairs amount to improvements, whether or not any fundamental changes in principle are adopted. For years past it has happened in many such cases that toothed gearing has been replaced by belts or ropes with most successful results. The natural tendency is to attribute such success to the change in principle, rather than to the improvement in the quality of the work. A defective arrangement of toothed gearing may be replaced with advantage by a moderate one of belt gearing, but neither is to be compared to an excellent one in which ropes are used. Similarly, a defective arrangement of rope gearing is not to be compared with an excellent one of toothed gearing.

CHAPTER XXXIX.

ROPE GEARING.

Conditions of adhesion.—Ropes are used very largely to replace belts in almost all positions in which they are not required to move from a fast to a loose pulley. Equally with belts, ropes depend upon frictional adhesion for their action; but the form of the groove is designed to increase the effect of such action by wedging. The co-efficient of friction for a rope working upon an iron pulley is practically identical with that of a belt. If a rope were arranged to work upon a plain cylindrical pulley, such as is used for belting, the conditions affecting adhesion would be the same, and the slack side of the rope would require to work at a tension of one-half that of the tight side, subject to the same correction on account of centrifugal force. If, however, the rope is arranged to work in a groove of the usual description, with the two sides inclined at an angle of 45° to each other in section, as in Fig. 121, the rope presses upon each side with a force about 30 per cent. greater than that with which it would press on a line upon the plain cylindrical surface, the rope being assumed to exist under the same tension at the part in question. This is equivalent to a direct increase in the co-efficient

of friction. Assuming the safe co-efficient of ·22, the potential co-efficient becomes $\cdot 22 \times 2{\cdot}613 = \cdot 575$. By this means a rope works with the slack side under a tension of one-sixth of that upon the tight side, under



A	B	C	D	E	F
DIMENSIONS IN INCHES					
1	3/4	1	1 5/16	3/16	1 1/2
1 1/4	15/16	1 3/16	1 5/8	1/4	1 7/8
1 1/2	1 1/8	1 3/8	1 15/16	5/16	2 1/4
1 3/4	1 5/16	1 9/16	2 1/4	3/8	2 5/8
2	1 1/2	1 3/4	2 9/16	7/16	3

Fig. 121.—Section of rope pulley, with grooves of usual angle = 45°.

conditions exactly equivalent to those of a belt with the slack side, under a tension of one-half that upon the tight side. In both cases the co-efficient in actual work may be relied upon to exceed ·28, and in both cases pure slipping cannot take place until the co-efficient as calculated upon a plain cylindrical surface

falls below ·22. In both cases also a slight creep takes place, the amount of which is probably a little greater in ropes than in belts, because of the greater difference in the tension under which the rope exists on the two sides. The creeping action takes place over the whole of the surface in contact, and causes the absorption of a small amount of power. A further amount of power is consumed in alternately placing the ropes in the grooves of the two pulleys and withdrawing them. But the amount of power accounted for in these ways cannot be very serious, as it seldom shows itself by causing serious heating of the pulley rims. It is usually estimated at from 5 to 10 per cent. upon the gross indicated horse-power of an engine, from which the whole power is taken off by ropes. But it is generally held that this loss is amply justified by the advantages secured in its judicious adoption.

Angle of groove.—The angle included between the two sides of the groove is a most important factor in the efficiency of the system. A certain amount of wedging action is necessary to secure the advantages referred to. But beyond this, any increase in the acuteness of the angle leads to excessive loss of power in the entry and withdrawal of the rope. An excessively acute groove also causes a greater difference in the working radii between different ropes, when slight variations unfortunately exist in the fit of the ropes in the grooves. Mr. James Combe of Belfast, who originated the use of rope gearing, made a number of experiments upon fixed pulleys, in which grooves of different angles were turned. Ropes were passed over these grooves and loaded differently at the two ends. Grooves of 45° were found to give amply secure adhesion without excessive bite in the groove. This angle was therefore ultimately adopted for the first

installation of rope gearing, which was erected about 1860 to drive 200 indicated horse-power, in the works of Mr. Combe's firm at Belfast. This drive is still at work, and is in all essential respects the same as would be erected under the same conditions at the present time. For single ropes, for special purposes, a groove of about 30° is sometimes adopted, as for instance in connection with ropes for working overhead cranes from below by hand and at slow speeds. In such cases the only tension which can be practically applied to the slack side is that due to the weight of the rope itself.

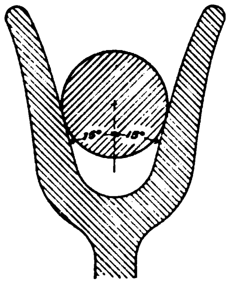


Fig. 122.—Section of rope pulley, groove = 30°.

A groove for this purpose is shown in Fig. 122, flaired to a greater width at the edge to avoid chafing the rope or running it off the pulley, when not led or guided quite accurately.

Grooves of guide-pulleys.—Pulleys whose office is the simple guidance of a steady rope are usually made with a plain semi-circular groove; or of a shape more or less approaching to the last, if the rope is liable to arrive at the pulley from variable directions. In either case the groove is usually made of such width as to allow the rope to lie on the metal at the bottom of the groove, and thus avoid slight exposure to wear, due to wedging

in the groove. This is quite successful when the pulley is required to effect a change in the direction of the rope, so that it lies in contact with the pulley over an arc sufficiently large to ensure such a drag as to keep the pulley in full motion. Failing this, a much better practice is to use the ordinary groove of 45°, and arrange the rope in contact on the two sides. When a guide-pulley is allowed to lag, or to stop entirely, the rope is worn very much more than can possibly arise from working in a good wedging groove, and the pulley itself becomes worn into flats.

Description of ropes.—Ropes in connection with ordinary work are described by the number of inches in circumference. But for gearing ropes, it is more convenient to use the diameter of the circumscribing circle. As the two classes of ropes possess very different qualities, there is no necessity for adhering to one method of measurement, but rather otherwise. The first ropes used by Mr. Combe were made of leather, cut from the hide in the longest possible strips and made up into a circular rope. These were deficient in uniformity; they were liable to become untwisted; and generally they were not successful. Afterwards manilla hemp was tried, and found so successful that its use has continued to the present time, while probably more ropes have been made from this material than from all others combined. For many years past cotton, of long and strong fibre, has been largely and increasingly used. Cotton is rather stronger, more durable, and more flexible than manilla, but is more costly in probably an equal proportion. The merits of the two are, however, so nearly balanced that the choice is decided upon local considerations rather than upon intrinsic merit.

Power of ropes.—In deciding upon the proportions to

be adopted in designing an installation of rope gearing, the same factors are to be regarded as in belt gearing. The tensile strength of a good manilla hemp rope, one inch in diameter, broken by ordinary means through the solid part, and without specially prolonging the operation, may be taken at 6000 pounds. That of the splice, however, does not exceed 4000 pounds when broken in the same way, while a much smaller load applied a very large number of times, as in ordinary work, will suffice to break the rope. In practice it is found to be very desirable to avoid the imposition of a greater load than 200 pounds upon such a rope. Larger ropes may be loaded in proportion to the sectional area, or to the square of the diameter. As in belt gearing, the total stress includes the primary stress due to the work performed, the secondary stress to balance the tension upon the slack side, and the tension due to centrifugal force. The tension due to centrifugal force $= \dfrac{\text{weight per lineal foot} \times (\text{velocity in feet per second})^2}{32{\cdot}2}$.

At low speeds this is very trifling. At all speeds it is less with light ropes than with heavier ones, which may be used for the same work. The tension due to centrifugal force is to be subtracted from the total permissible stress, so as to leave the sum of the primary and secondary stresses. As an example, a rope may be taken, $1\frac{1}{2}$ inches in diameter, working at 4,500 feet per minute. This rope may be loaded to 450 pounds and will weigh ·74 pounds per lineal foot. The tension due to centrifugal force will be 129·3 pounds, leaving 320·7 pounds as the working stress upon the tight side. One-sixth of this $=$ 53·4 pounds $=$ secondary stress, leaving 267·3 pounds as the primary stress, due to the work done. $\dfrac{267{\cdot}3 \times 4{,}500}{33{,}000} = 36{\cdot}4$ horse-power which may be transmitted by rope.

Table of powers normally transmitted by ropes.—In Table XXIX. the maximum tension upon each rope is assumed to be 200 pounds per circular inch, the tension upon the two sides are taken in the proportion 1 : 6, after the elimination of centrifugal force, contact is allowed upon an arc of 180°, or half of an entire circle, and the diameter of the least pulley is assumed to be not less than forty times greater than the diameter of the rope. The heads of columns give the diameters of ropes, the loads or tensions, and the weights of ropes per lineal foot. The columns marked C.F. give the tensions in pounds due to centrifugal force; and those marked H.P. give the respective horse-powers which may be transmitted under the conditions. If it were possible to abolish the secondary stress, and work the slack side of the rope subject only to the tension due to centrifugal force, the horse-power which might be transmitted in any particular case would be one-fifth greater than that given in the table.

Power as affected by speed of ropes.—It will be observed that the power transmitted by a rope increases rapidly with the speed, up to a certain point. Consequently, a high speed is very desirable, with a view to secure a low original cost in the installation, and a moderate amount of space occupied. On the other hand, when a speed of 4,500 feet per minute is exceeded, each increase of speed gives less and less increase in power, and less advantage in respect to cost of plant, and space occupied. The speed should therefore be arranged between the limits of 3,300 and 4,500 feet per minute where possible. Higher speeds may be adopted for special reasons, such as securing a higher efficiency of fly-wheel, or avoiding the use of pulleys of diameter insufficient for the ropes adopted.

Influence of angular measure of arc of contact upon

pulleys.—When the arc of contact is reduced below 180°, the several ropes are unable to transmit so much power. Table XXX. gives the ratio of power which may be transmitted by a rope according to the angular

TABLE XXIX.—POWER TRANSMITTED BY GOOD ROPES OF COTTON OR MANILLA HEMP.

Maximum load, 200 pounds per circular inch. Contact, 180 degrees. Tension on two parts, 6 : 1. Results corrected for centrifugal force.

Diameter, 1 inch Total load, 200 lbs. Weight per foot, ·33 lbs.			1¼ 312·5 ·51		1½ 450 ·74		1¾ 612·5 1·00		2 inches. 800 lbs. 1·30 lbs.	
Velocity in feet per minute.	C.F.	H.P.	C.F.	H.P.	C.F.	H.P.	C.F.	H.P.	C.F.	H.P.
1,500	6·4	7·3	9·9	11·4	14·4	16·5	19·4	22·4	25·2	29·3
1,800	9·2	8·7	14·2	13·6	20·7	19·7	28·0	26·6	36·4	34·7
2,100	12·6	9·9	19·4	15·5	28·2	22·4	38·0	30·5	49·5	39·8
2,400	16·4	11·1	25·4	17·4	36·8	25·0	49·8	34·1	64·6	44 6
2,700	20·8	12·2	32·1	19·1	46·5	27·5	62·9	37·5	81·7	49·0
3,000	25·6	13·2	39·6	20·7	57·4	29·7	77·6	40·5	100·9	53·0
3,300	31·0	14·1	47·9	22·0	69·5	31·7	93·9	43·2	122·1	56·5
3,600	36·9	14·8	57·0	23·2	82·7	33·4	111·8	45·5	145·3	59·5
3,900	43·3	15·4	66·9	24·2	97·1	34·7	131·2	47·4	170·6	62·0
4,200	50·2	15·9	77·6	24·9	112·6	35·8	152·2	48·9	197·9	63·9
4,500	57·6	16·2	89·1	25·4	129·3	36·4	174·7	49·8	227·1	65·1
4,800	65·6	16·3	101·4	25·6	147·1	36·7	198·8	50·1	258·4	65·6
5,100	74·0	16·2	114·2	25·5	165·8	36·6	224·0	50·0	291·2	65·5
5,400	83·0	16·0	128·3	25·1	186·1	36·0	251·5	49·5	327·0	64·5
5,700	92·5	15·5	143·0	24·4	207·4	34·9	280·3	47·8	364·4	62·7
6,000	102·5	14·8	158·4	23·3	229·8	33·4	310·6	45·7	403·8	60·0

measure of the arc of contact. Thus, when the arc of contact measures only 120°, any particular rope will only transmit 84 per cent. of the power of which it is capable when the arc of contact measures 180°. In both

cases, the rope will be loaded to the same degree, the tension due to centrifugal force is the same, and there is no appreciable difference in the power lost by resistances. The difference, however, arises from a higher secondary stress, necessary to prevent slipping. A rope or belt drives better when the tight side is beneath, chiefly because the deflection of the slack side causes an increased arc of contact upon the pulleys.

TABLE XXX.—VARIATION IN POWER TRANSMITTED BY ROPES ACCORDING TO VARIATION IN ANGLE OF CONTACT.

Angle subtended by arc of contact of rope.	Ratio of tension, on two parts of rope with constant co-efficient of friction.	Ratio of horse-power which may be transmitted by same rope.
180°	6·00	1·00
165°	5·17	·97
150°	4·45	·93
135°	3·83	·89
120°	3·30	·84
105°	2·84	·78
90°	2·45	·71
75°	2·11	·63
60°	1·82	·54
45°	1·57	·44
30°	1·35	·31
15°	1·16	·16
0°	1·00	·00

Comparison of different ropes and belts with respect to space occupied.—When the calculations in connection with a general design have been made upon the basis of the adoption of such a diameter of rope as would be most suitable for the power, speed, and diameter of pulleys, it may be found impossible to provide width for the fly-wheel and line-shaft pulleys, and for means of

access to them. In such a case, the adoption of thicker ropes may give the required relief, but if these are more than one-fortieth part of the diameter of the smallest pulley upon which they work, they must be expected to wear away more rapidly than usual. Such excessive wear may be mitigated somewhat by the adoption of liberal proportions, but in the case assumed this is probably quite impossible. Table XXXI. shows the horse-power transmitted by ropes of various diameters, working at a

TABLE XXXI.—COMPARISON OF POWER TRANSMITTED IN RELATION TO WIDTH.

	Distance apart centre to centre of grooves.	Horse-power.		Pressure upon bearings in pounds.	
		Per rope.	Per inch in total width.	Per rope.	Per horse-power.
Cotton ropes, working at 4,500 feet per minute.					
1 inch dia.	1½ inches	16·2	10·8	166	10·3
1¼ do.	1⅞ do.	25·4	13·5	260	10·3
1½ do.	2¼ do.	36·4	16·2	372	10·3
1¾ do.	2⅝ do.	49·8	18·9	509	10·3
2 do.	3 do.	65·1	21·6	668	10·3
Square leather ropes, working at 4,500 feet per minute.					
1 inch sq.	2 inches	30·1	15·0	515	17·1
1½ do.	3 do.	62·6	20·9	1,071	17·1
2 do.	4 do.	104·0	26·0	1,780	17·1
Belts working at 4,500 feet per minute.				per inch width.	
One ordinary double belt ...			9·3	204	22·0
Two do. superposed			16·7	367	22·0
One superior double belt ...			10·8	238	22·0
Two do. superposed			19·6	430	22·0
Chain belting, one inch thick ...			6·3	139	22·0
Toothed gearing, working at 2,250 feet per minute, minimum pressure on bearings					14·7

In each case independent of pressure upon bearings due to weight of wheels, shafts, or other details.

speed of 4,500 feet per minute. Also the horse-power per inch in width of fly-wheel or pulley. The powers per inch in width transmitted by several classes of belting are also given, from which it will be seen that all ropes above one inch in diameter possess in this respect a great advantage over a single belt of double thickness, ordinary quality. But a superposed belt gives the means of increase to a greater power per inch than that transmitted by ropes of 1½ inches diameter. Two belts of superior quality, one superposed upon the other, will transmit almost as much power per inch as ropes of 1¾ inches diameter. In such comparisons, the ultimate advantage in short drives will generally rest with belts on account of the facility for accurate adjustment to length, and for long drives with ropes, on account of moderate cost.

Distance of centres.—In all cases it is better to avoid, if possible, belt or rope transmissions in which the centres are less than 30 feet apart, measured in a horizontal direction. As already explained in connection with belts, the horse-power transmitted is also subject to a great reduction, on account of the weight of the rope, when working to a great height. A rope 1½ inches in diameter, connecting two shafts at about the same level and a moderate distance apart, working at a speed of 4,500 feet per minute, will transmit 36·4 horse-power. But if the shafts are at a vertical distance of 100 feet apart, the power transmitted will be only 28·0, or 23 per cent. less. This arises from the fact that 100 feet of rope weigh 74 pounds, which amount of additional tension is imposed upon the rope near the upper pulley, and must be allowed for in strength.

Comparison as to pressure upon bearings.—Table XXXI. also shows the actual pressure upon shaft bearings due to the sum of the primary and secondary stresses. In

rope gearing, where the total working tension upon the tight side is six times as great as that upon the slack side, the pressure upon the bearings is 10·3 pounds per horse-power, at a speed of 4,500 feet per minute, and varies inversely as the speed. In belt gearing working at the same speed, with tensions in the proportions of 2 to 1, the pressure upon the bearings is 22·0 pounds per horse-power. At all speeds, the ratio of advantage of ropes over belts in this respect remains constant. In many cases, the adoption of belts or ropes involves such a great width of fly-wheel pulley, that the fly-wheel shaft—usually the crank-shaft of the engine—deflects to a very serious extent. This subject, and measures to be adopted in anticipation, are discussed in another chapter.

Disadvantages of ropes of large size.—Ropes above 2 inches in diameter should never be used for ordinary driving, on account of their deficient flexibility, which leads to loss of power and excessive wear of rope. In no case should the rope be greater in diameter than one-thirtieth part of the diameter of the smallest rope pulley, and if the diameter of rope exceeds one-fortieth part of that of the pulley, some reduction should be made in the power transmitted. In all cases, the ropes should be as small in diameter as the conditions of the case will admit. At the present time the general tendency appears to be towards the use of ropes of about 1½ inches diameter.

Square ropes of leather.—Ropes of square section are made by Messrs. Tullis from orange tan leather, which is notable both for strength and lightness. Each rope is precisely equivalent to an ordinary leather belt, with sufficiently numerous plies superposed to make the total thickness equal to the width. Each rope is set to run in a separate groove, whose sides are inclined

at 45° or at 90° to each other, as in Fig. 123. The entire rope is cemented together between the plies and in the joints, which are of the usual scarfed type, arranged to break joint. Ropes of 1 inch, 1½ inches, and 2 inches square may be loaded to maximum tensions of 450, 950, and 1,600 pounds respectively, the larger sizes being rather less severely loaded, in proportion to sectional area. Considerably less loads should, however, be imposed where possible, and imperatively so if for any reason it is not found practicable to make the closing joint quite as good as the rest. These ropes

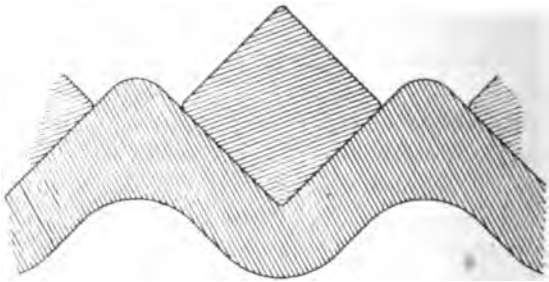


Fig. 123.—Section of square leather ropes and grooves.

weigh about ·47 pounds per lineal foot, per sectional inch in area. Independently of the tension due to centrifugal force, the tension on the tight side may be 2½ times as great as that upon the slack side.

The advantages of these ropes consist in their ability to transmit greater power per inch of width than either textile ropes, or ordinary belts, except where not less than three double belts are superposed. Also in their superior facility of adjustment as to length, which is of greatest value where the centres of shafts are near together.

Distribution of power by means of ropes.—The usual manner of applying ropes for heavy driving is to provide

a wide fly-wheel, turned in grooves, to receive a sufficient number of ropes for the transmission of the required power. These ropes are distributed to the several shafts which require to be driven, as in the case of belts, with the exception that, according to the amount of power required to be applied to each shaft, the number of ropes is varied instead of the width of the belt. The grooves should be spaced at about the distances apart given in Table XXXI. and p. 707. For special reasons these distances may be slightly reduced, but they should be sufficiently wide for the ropes to work well clear of the sides of the grooves, and to allow for a little excess in diameter in new ropes or splices.

Importance of uniformity of ropes.—In order to ensure that each rope shall take its fair share of work, it is essential that the centre of each, when in the groove, shall lie at the same distance from the centre of the shaft. The first condition towards securing this object is that the whole of the ropes shall be of the same diameter originally, and shall be equally affected by wear. This can only be relied upon when all the ropes are made together. By this means also the further advantage of uniform elasticity is secured.

Importance of uniform accuracy of grooves, and of verification of same.—The grooves in each pulley must be cut with accuracy, so that the rope in one groove will not sink deeper than that in another groove. In order to check this point, the circumference of the flange at each side of the pulley should be measured by means of a steel tape or wire. Each flange should be of absolutely the same diameter, so that a straight-edge laid across the two would give a line exactly parallel to the centre line of the shaft upon which the pulley is mounted. A template cut to accurately fit one groove, and marked across along the straight-edge, should fit all others with

equal accuracy, and with the mark exactly on the line of the straight-edge. If in testing a wheel which is erected in position, one flange is found to be larger in diameter than the opposite one, the straight-edge should be brought parallel to the shaft, just as though both flanges were of the same diameter. With a wooden straight-edge this may very easily be done, by the insertion and adjustment of an ordinary wood screw to rest upon the flange which is of smallest diameter. If upon each pulley all the grooves conform to this test, and the ropes are uniform, they will work successfully at a fairly uniform tension. But if in the driving pulley any groove is tighter than the rest, the rope working in it will run at a greater radius, and will therefore run on to the pulley at a correspondingly higher speed than the rest. Assuming that in such a case the grooves upon the driven pulleys are uniform, the rope cannot leave that pulley at any increased speed without slipping or stretching. It therefore suffers alternate stretching upon the tight side and relaxation upon the slack side, which causes excessive wear and tear upon the rope. This proceeds until the rope assumes a smaller diameter, and sinks towards its correct position in the groove of the driving pulley. But while this takes place, the rope must also sink more deeply in the groove of the driven pulley. When—as is usually the case—the pulleys are of different diameters the rope will be still further overloaded, as a result of wear. It is barely possible that inaccuracies in two pulleys will correct each other, but this is most improbable, and therefore every care should be taken to avoid inaccuracies of all kinds.

When a rope-driving arrangement is in satisfactory operation, all the ropes which work together will be observed to hang with equal tightness on the tight

side. A comparatively large variation upon the slack side may arise from variation in the spliced length of the several ropes without serious trouble, though if this side happens to be beneath, the arc of contact will be reduced.

Uniformity of groove is in some cases secured by the use of a milling tool, the spindle of which lies in a direction perpendicular to the general surface of the pulley. In some other cases, a strong cutting tool of constant form to finish the groove is used, after the whole have been roughly turned to form. Each groove should be finished to a dead smooth surface, but not polished.

Multiple grooves for separate ropes.—In ordinary British practice, a separate rope is provided to fit only one groove in each of two pulleys. Each rope is spliced together to the correct length for the purpose. The splicing requires to be most carefully performed, so as to avoid on the one hand excessive size of splicing, and on the other hand an inordinate sacrifice of strength. Usually a successful compromise between these conditions is effected by the exercise of greater skill than is necessary in joining a belt. The splicing is performed with the rope removed from one or both pulleys, and if possible the operation should be performed in daylight. The attainment of the required degree of accuracy in respect to length is especially important, and difficult when the centres are very close together, so that there is a comparatively short length of rope to supply elasticity. As a rule, the adoption of belts is to be recommended in such cases, on account of the possibility of greater accuracy in jointing, though belts also work less efficiently when connecting shafts at a shorter distance than 30 feet apart. A short rope may be adjusted or tightened by means of a movable guide-pulley. This

means is equally applicable to a number of ropes working together, and which are equally tight. Independent adjustment of each rope would, however, introduce an objectionable amount of complication in the details.

Driving by multiple parts of a single rope.—The difficulty experienced in connection with the adjustment of a series of separate ropes, especially when of comparatively short length, has led to the use of one long rope, running successively over all the grooves and returned from the last to the first groove, by means of a guide-pulley or tension-carriage provided with means of adjustment, to give the required amount of tension to the rope. When the connected shafts are very close together, as when an engine and dynamo placed upon one bed-plate are so connected, this is the only successful way in which rope transmission can be adopted. It has been largely used for large and small transmissions in the United States, and is sometimes called the "American system," though it is believed that it was originally designed by Messrs. Combe. In connection with its adoption, strict uniformity in the several grooves of each pulley is originally of equal importance as in an installation in which separate ropes are employed. But it is obvious that the whole of the rope must wear with practical uniformity, so that the irregularity in the work done by the several parts does not increase by reason of such wear.

Tension affected by stretch.—On account of the uncertain stretch to which all ropes and belts are subject in work, it is very difficult to ensure that an accurately adjusted amount of tension shall at all times be imposed upon them. Hence it often arises that they are exposed to excessive tension, whereby they suffer damage, or the tension becomes deficient, so as to cause slip.

Deflection as evidence of amount of tension upon ropes and belts.—Ropes which connect pulleys at nearly the same level, which weigh ·33 pounds per lineal foot per circular inch, may be exposed to a total load of 200 pounds per circular inch. The tight side of such ropes when at work will approximately follow a curve of 600 feet radius, of which the versine or departure from a straight line in feet $= \left(\frac{\text{length or chord in feet}}{69{\cdot}3}\right)^2$. In like manner, double belts of 3 pounds per square foot may be loaded to 180 pounds per inch in width, when the versine $= \left(\frac{\text{length or chord in feet}}{75{\cdot}9}\right)^2$ in feet.

Tension measured by spring balance.—A spring balance may be used to ascertain the tension upon each rope, by hauling upon until the required tension is obtained, then recording the curve at a sufficient number of points and comparing with that followed when at work. The curve for each rope will be the same, so that it is not necessary to test each rope. Such a test should be applied near to the upper end of the driving side. The spring balance should pull in line with the rope, and the rope should be slack between the point of application and the upper pulley, to ensure that the balance receives the whole of the pull. For this reason also the rope should be clamped to the upper pulley to prevent slipping round. Belts may be tested in the same way, but wide belts will require a stronger balance than ropes, on account of the greater tension.

Crossed ropes.—Ropes may be applied in the same way as crossed belts, to give opposite rotation. If however, in such cases, more than a single rope is adopted, the centres of the several grooves must be spaced at distances about equal to 2½ diameters of the rope, to give room for crossing. In such a case the

ropes do not appear to suffer chafing more than ordinary crossed belts. But the suspicion of chafing would be avoided by throwing the shafts out of parallel, while approximately maintaining a constant distance apart, when this can be done.

Possible deviation of ropes and the use of exceptional grooves.—A belt must leave each pulley at a point in the plane of the pulley with which it next comes into contact, subject to slight modification by reason of variable diameter of pulley, or by a stationary guide or roller for the purpose of preventing the belt from running off the pulley. The same principle applies to the

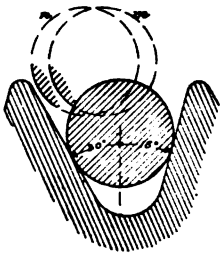


Fig. 124.—Section of non-symmetrical groove.

absolutely correct working of ropes. But with these there is some considerable amount of freedom admissible, so that it is possible to lead a rope on to the pulley, from directions differing widely, so long as the departure is not so great as to bring the rope in contact with either edge of the groove. As a measure of relief in such cases, the groove may be canted a little to one side, so long as the angle included between the two sides of the groove remains constant. In Fig. 124 a groove is shown of which one side stands at 30° from the centre plane, and the opposite one at 15°, so that the two make 45° as in the ordinary groove.

Fig. 125 shows a groove in which the respective angles are 40° and 5°. One side might be vertical, but this is almost sure to lead to chafing of the rope. In extreme cases, where the rope leads on from one direction and

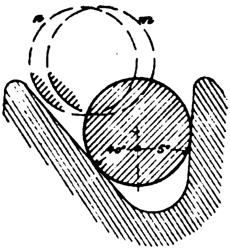


Fig. 125.—Section of non-symmetrical groove.

off towards the opposite direction, the groove may be made to a wider angle, but this involves a sacrifice of driving power, owing to the diminished wedging action, and consequent necessity for greater tension upon the slack side. Fig. 126 shows a groove of

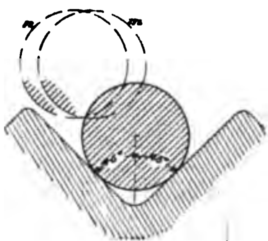


Fig. 126.—Section of pulley-groove = 90°.

90°, or 45° on each side of the centre plane. In a groove of this form any given rope will only transmit 75 per cent. of the power which it will transmit in a groove of 45°. Similarly, a rope working in a symmetrical groove of 60° degrees will only transmit 90

per cent. of the power which it would transmit with equal ease in a groove of 45°.

In all cases the angular ratio of possible departure from the mean position must be tested by geometrical construction, or on correct models, or calculated. Chafing must be avoided, and sliding reduced to a minimum. The maximum angle of departure varies, within ordinary limits, inversely as the square root of the diameter of the pulley, *i.e.* the departure may be greater with a small pulley than with a larger pulley, in which a groove of the same section is adopted.

TABLE XXXII.—OBLIQUITY OF ROPE GROOVES AND ROPES.

	Reference to illustration.	Angle of side of groove measured from central plane.	Diameter of pulleys. 5 feet.	Diameter of pulleys. 20 feet.
			Departures of ropes from central plane of pulleys.	
Allowing rope to run one-quarter inch clear from side of groove on entering and leaving	Fig. 124, *m*.	30°	1 in 30	1 in 60
	Fig. 125, *m*.	40°	1 in 15	1 in 30
	Fig. 126, *m*.	45°	1 in 12	1 in 24
Allowing rope to run just clear from side of groove on entering and leaving	Fig. 124, *n*.	30°	1 in 15	1 in 30
	Fig. 125, *n*.	40°	1 in 10	1 in 20
	Fig. 126, *n*.	45°	1 in 9	1 in 18

Figs. 124, 125, and 126 each show two circles, *m* and *n*, dotted in the positions of ropes leaving the groove. Table XXXII. shows the angular ratios applying to these in several cases. In each case no allowance

is made on account of the curve arising from the weight of the rope, which is small and variable in amount, according to the direction of the rope. Also no allowance is made on account of oscillations of ropes which arise in work.

The application of these principles to cases of oblique transmission, corresponding to quarter-twisted belts and guide-pulleys, is obvious. In quarter-twist work, ropes may be used singly, or a single rope may be used over a series of grooves, or a series of separate ropes may be

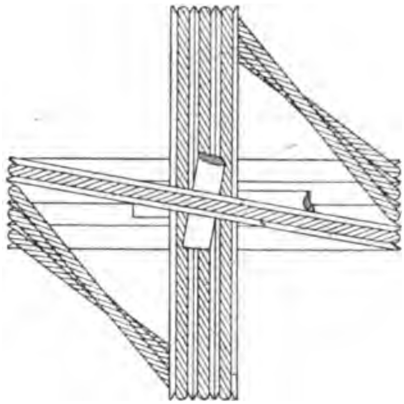


Fig. 127.—Oblique transmission by rope.

used in separate grooves. A typical case is shown in Fig. 127, in which a single rope is adopted to work over three pulley-grooves on one shaft, and four on another shaft, while a tension pulley is mounted separately, so as to maintain uniform tension in the system. The angle of each groove is 60°, or 30° on each side of the central plane. In such cases a compromise is often necessary, such, for instance, as a reduction in the diameter of each pulley below what is desirable in ordinary work, the object of such reduction being to

diminish the obliquity of the rope, and so relieve one point, though at the expense of another. In Fig. 127, the pulleys are shown 4 feet in diameter, measured over the centres of the ropes, the rope is 1½ inches in diameter, or one thirty-second part of the diameter of pulley, and the greatest obliquity is 2 feet 3 inches. Table XXXII. gives the possible obliquity or departure of the ropes, allowing slight clearance, to be 1 in 15 upon a pulley 5 feet in diameter, with a groove of 30° from the central plane. This corresponds to an angle of 1 in 13½ for pulleys of 4 feet diameter. Consequently, the two shafts shown in Fig. 127 should be placed not less than 30 feet apart, to work satisfactorily at a moderate speed, say 150 revolutions per minute. If the work is very uniform and the speed slow, the rope may be allowed to rub the flange slightly, at the sacrifice of durability, but no rule can be given to govern such cases. It is obvious that in all cases involving obliquity of ropes, the smallest practicable diameter of rope should be adopted, with the direct object of reducing the diameter of pulleys, and thereby the amount of obliquity. But small ropes are limited in working power, and all ropes are placed at a disadvantage, when they are used upon pulleys of excessively small diameter, as compared with others upon pulleys of larger diameter running at the same number of revolutions. Such ropes should be underloaded, or great loss of power will be incurred by reason of an excessively tight condition which becomes necessary, and which also leads to excessive cost on account of maintenance and renewals.

Cleanliness of groove surfaces, and balance of pulleys.—The working surfaces of all rope grooves should be kept quite clean. The ropes themselves will preserve the metal in a clean condition along the lines of contact.

But if the grooves are allowed to fill with rubbish beneath the ropes, it is likely to drop out and cause mischief. All pulleys should be well balanced, and though this may have been originally attended to, the advantages may be lost by want of attention to cleanliness.

Physical condition of ropes.—Ropes should be treated with castor oil to such an extent as to secure a soft and supple condition. The dressing with oil should be repeated when necessary, but such an excess of oil as to show a film upon the pulley should be scrupulously avoided. This leads to slipping and also to weakness of the rope itself, as it is well known that wetness, tar, or grease causes a loss of holding power between the separate fibres of the rope. Mineral oil is not beneficial to a rope, as is castor oil; and a rope heavily dressed with such oil should be avoided.

Ropes should not be largely exposed to the weather or damp. They suffer from such exposure to a much less extent than leather belts, but it should be avoided where possible. A small jet of steam may be used with advantage to humidify the air in the belt-race, but when the ropes are in good condition this should be unnecessary.

Ropes may be subjected to dissection, and examination made of the materials entering into their composition, with a view to comparison of different samples, in the same way as woven belts.

Periodical examination of ropes.—Ropes in use should be subjected to periodical examination and report, precisely in the manner of belts. Very often a light iron plate or board is mounted upon hinges, in such a way as to move and cause a bell to ring when the former is struck by any loose end which may become disengaged from the splice of a rope.

Protection against chafe.—Serving, or small yarn closely coiled, is sometimes applied to a rope as a protection against wear. This should be preceded by a worming of yarn laid between the strands, so as to make up an approximately circular form, and thereby avoid cutting the serving by wear along the strands of the rope. A parcelling of canvas should also be applied beneath the serving as an additional means of elasticity. Serving increases the diameter but not the strength of a rope, so that wider grooves are necessary, and more space is occupied. It is a necessity in the rare cases in which wire rope is adopted, but otherwise is of doubtful utility.

General conditions of application.—Rope gearing arrangements are designed in a manner very similar to those in which belts are the leading feature. Both are more silent in work than toothed gearing, and less liable to serious breakdown and interruptions to work. Generally speaking, the original cost of ropes is much less than that of leather belts, and rather less than that of woven belts. Leather belts last longer than ropes or woven belts under ordinary working conditions, but in damp situations or warm climates, or under other exceptional conditions, the order may be reversed. The grooved pulleys for ropes cost more than the plain ones for belts. The absolute pressure upon the bearings due to the power transmitted, apart from that due to weight of parts, is given in Table XXXI., which refers to ropes and belts working at 4,500 feet per minute, and to toothed gearing working at 2,250 feet per minute. In this respect, the same relative condition prevails at other speeds, provided that the two former are compared at the same speed with the latter at one-half the speed. The pressure upon the bearings is greatest with belts, but its importance is very much less at the present

time than it was before the introduction of improved means of lubrication. Ropes and belts allow an amount of endlong movement in the shafts which is impossible with toothed gearing, and which causes the production and maintenance of excellent surfaces in the bearings.

CHAPTER XL.

PULLEYS.

Tension due to centrifugal force.—Pulleys for belt or rope driving require to possess sufficient strength in the rim to withstand the physical rough usage to which they are exposed in transit and erection. When this is the case, the strength is more than sufficient for the driving power. The tension due to centrifugal force at any given speed increases in proportion to the sectional area of material in the rim, so that the tension per square inch due to this cause is the same upon a heavy as upon a light rim. The tensile stress in pounds per square inch upon any cross-section of a uniform pulley rim, due to centrifugal force =

$$\frac{\text{Weight of rim in pounds per lineal foot} \times (\text{velocity in feet per second})^2}{32 \cdot 2}.$$

Allowing a tension of one half-ton per square inch upon a rim of cast-iron, the speed may reach 6,440 feet per minute, so that cast-iron of uniform section and good quality may be used for all speeds which are practicable in other respects.

The tension due to centrifugal force is imposed as a direct tensile stress upon the uniform rim, and upon each joint. The tensile strength of the joint should therefore be calculated in detail, as to its sufficiency for

the purpose. The weight of any flanges or irregularities which may exist in any position between the arms must also be regarded. In the exact calculation of this item, allowance must be made for the application of the centrifugal force due to such irregularities as a transverse load upon the rim, which is primarily in a state of uniform tension. But a sufficiently close approximation will be obtained by assuming two to four times the additional weight to be distributed over the distance between two arms, and adopting the combined weight in the above calculation.

Surfaces.—The grooves of rope pulleys are treated in the chapter on rope gearing. The cross curvature of belt pulleys is also treated in the chapter on belt gearing. All edges should be moderately rounded to prevent accidents.

Arms.—The arms of pulleys must possess sufficient strength to act as cantilevers to transmit the power from the boss to the rim—or the opposite. The cross-section of the arms may be either oval or cruciform. On the whole the former is best, in opposing least air-resistance to motion. For the sake of strength and appearance, a moderate amount of taper should be allowed. The centre line may be either straight and radially disposed, or curved in various ways. Curved arms are usually adopted with the intention to allow yielding during contraction in the sand, and so reduce the strain to which the pulley as a whole is subjected during the process. It is, however, well established that good castings may be made in either way, by the adoption of good proportions. As in the case of toothed wheels, arms are often replaced by a continuous plate cast solid with the rest of the pulley.

Boss.—The boss of a pulley requires to be sufficiently strong to give the requisite pressure upon the keys, as

dealt with in connection with wheels. Conical keys are used more largely in connection with pulleys than with wheels. In these the pressure is applied throughout the circumference. Allowing a nett stress of one ton per square inch of sectional area, a co-efficient of friction of ·10, and a working tension upon the belt of 60 pounds per inch in width of belt, the minimum combined sectional area in square inches of the two sides of the boss =

$$\frac{\text{Diameter of pulley in inches} \times \text{width of single belt in inches}}{\text{Diameter of shaft in inches} \times 11{\cdot}7}.$$

This will just suffice with careful driving of the keys, and any desired allowance of surplus strength may be

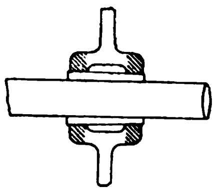


Fig. 128.—Boss of pulley, with conical keys.

made. In many cases the centre of the boss is chambered, as in Fig. 128, when only the sectional area which is cross-hatched, is directly supported by the keys, and is available for strength against bursting, though the chambering imparts width and stability to the attachment. The bosses of pulleys of all dimensions are usually made of greater thickness than any other part of the pulley. When the cooling of the boss in the mould is not accelerated, it remains in a fluid condition longer than the rim and arms; therefore the contraction of the latter is completed before that of the former, and one or more arms are likely to be drawn apart. To avoid this risk the boss is often divided

into three or four parts in the mould, so that the contraction takes place over the spaces thus formed. Such a boss is obviously worthless for keying against, until reinforced by wrought-iron rings or hoops, which are shrunk against shoulders prepared for them. The split openings are filled tightly with iron plates before the hoops are shrunk on. It is, however, better to cast the pulley whole, in such a manner as to avoid any serious risk of breakage. Hoops may still be added with advantage to solid bosses.

Wrought-iron pulleys. — Large numbers of belt pulleys are made of wrought-iron throughout, or with the exception of a cast-steel boss. These are of less weight than cast-iron pulleys, and are consequently largely adopted in cases in which the cost of transport is great. They are deficient in stiffness, and often run with great inaccuracy. They, however, can be made in halves with great facility. Large pulleys for ropes or belts are often made with a cast boss, a cast or wrought rim, and a series of arms of rolled plates, angles, or T-bars. In many cases a better arrangement would be to adopt a cast boss and rim, connected by a plate at each side, of diameter equal to the turned diameter inside the rim. Such a plate would be fitted to the boss upon a turned surface, and secured by turned set-screws. The rim would be provided with two flanges turned on their outer faces for the plates to bed against, and the two plates drawn together by a series of bolts, so that the drive is by means of frictional contact. The minimum number of bolts would be the same for all pulleys of equal power and speed along the line of bolts, whatever the diameter of each pulley. In some cases angles or T-bars may be used to stiffen the plates, and in others similar plates may be used as centres for the attachment of separate arms.

Fast and loose pulleys.—In an ordinary pair of fast and loose pulleys, the former is secured upon the shaft by a key or a set-screw, or both. The loose pulley is sometimes arranged to run upon the bare shaft. In others it is fitted with a cast-iron bush, which may be bored to fit the shaft either at its original or a reduced

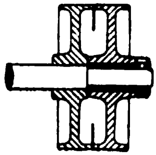


Fig. 129.—Fast and loose pulleys, ordinary pattern.

diameter, as in Fig. 129. By the use of the bush, two cast-iron surfaces are brought into contact, and greater durability secured than when cast- and wrought-iron work together; the bearing area is increased; and the bush is easily replaced by one of any diameter, after

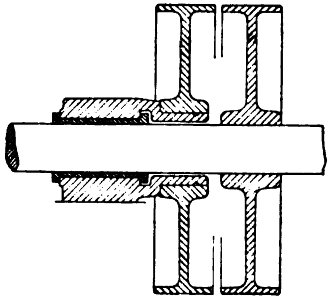


Fig. 130.—Fast and loose pulleys, standing drum arrangement.

wear. The whole should be so proportioned that the rims cannot come into contact. The fast pulley of such a pair is always the driven pulley, and the belt can only be stopped by running off the pulleys. In some cases this incessant work leads to great wear and tear of the belt and loose pulley.

Standing drum.—An arrangement is often adopted whereby the belt is always at rest when on the loose pulley, and the latter never moves except during the transfer of the belt from one pulley to the other. The arrangement is shown in section in Fig. 130. Any ordinary pedestal is used, fitted with brasses. To the body of the pedestal is attached a half bush, and a corresponding half to the cap. Arrangements should be made for easy access to the bearing. Very great advantage is derived from the use of such an arrangement in connection with heavy machines, though the belt requires to be assisted round by hand in starting, and possibly in stopping.

CHAPTER XLI.

LINE-SHAFTING, COUPLINGS, AND BEARINGS.

Material and connection of shafting.—Line-shafting of large sizes is usually made of iron or steel, by forging in the same way as crank-shafts. Shafts of diameters not exceeding 4 inches are made from rolled bars of iron or steel, turned to gauge in a lathe, and polished except where covered by wheels, couplings, or other details. Steel rolled bars can be obtained up to 6 inches diameter, but when collars or bosses are required, it is best to make or obtain an iron or steel forging ready for the lathe. Hot- and cold-rolled or planished shafting sufficiently smooth and circular for use without turning is employed to a limited extent where only plain parallel lengths are required. Long lengths of shafting are usually made in pieces of from 20 to 30 feet, for facility in erection, and in making alterations. Formerly, all shafting was provided with bosses at each coupling end, keyed and fitted. This plan gave great facility for inserting and removing keys without damage to the shaft surface; and by increasing the radius of frictional hold in the coupling, diminished the chances of the key or the fit of the coupling working loose. This practice is still followed by many good firms, but by others is abandoned in favour of shafts without

bosses, which are cheaper to produce, and allow the fixing of unsplit pulleys otherwise than by conical keys. Wrought-iron for shafts is rapidly disappearing in favour of steel, which is more uniform and free from veins. Steel is of greater strength than iron, and free from defects to which iron is subject, especially those on the surface, which cause iron to be less trustworthy as to absolute strength, and also more likely to wear, cut the bearings, and giving trouble by heating. For these reasons, steel shafting may be made of smaller diameters than iron, by which is also secured a proportionate saving in loss of power by friction. Steel is stiffer than iron of equal dimensions, but owing to the smaller diameters adopted, shafts of equal driving power are less stiff than in iron.

Plain coupling.—The plain muff, box or cylindrical coupling is still largely used, and for permanent work it combines the greatest security with a neat appearance, without projections of any kind. The two shaft ends are fitted in one coupling, and secured by keying. Bossed shaft-ends are therefore necessary for the insertion of the keys in these couplings. Each shaft-end should fit tightly in the coupling, and it is better if the latter is finished in the lathe when all keyed together. Usually, one sunk key of the entire length of the coupling is used; in other cases two short keys are used to follow each other. The latter course enables the keys to be more tightly driven, but with fairly good work the former arrangement gives no trouble. Some engineers make two shaft-ends in one piece to be fitted to the coupling, keyed, parted, and welded to the respective shafts. This is more costly than the usual way, and possesses no advantage of importance. If a bearing on each side of the coupling is necessary for the support of the shaft, the two bearing

surfaces should be finished in the lathe when the coupling is tightly keyed in position, or there is some risk of the two working slightly out of truth. Forty years ago, shafts were coupled by half-lap joints, but these were proved to possess no advantages, and were excessively costly to make. Shafting connected by plain couplings must be keyed together after the whole of the several lengths are lifted into the bearings. But in many cases the whole series may be moved longitudinally to some distance from its working position. When this cannot be done, and the whole or part of the keys cannot be driven in position, some other type of coupling must be adopted at the points affected, face-plate couplings being usually selected.

Face-plate couplings keyed on shafts.—Face-plate couplings are made in pairs, one piece or block being fixed on the end of each of the connected shafts. The couplings are bored to fit the shafts with accuracy, and secured by one—or occasionally two—keys, each fitted in a sunk groove, and driven in from the end when the shafts are apart. Great security is thus obtained, and also facility for removal in the reverse direction. The keys being inserted from the ends, the grooves may be sunk below the main diameter of the shaft, so that the necessity for raised bosses is avoided. The two parts of each pair of couplings are secured together by means of bolts. The bolt-holes are rimered together and the bolts tightly driven. As an additional means of security, one coupling is made with a spigot accurately fitting into a recess in the fellow coupling. Sometimes the same object is achieved—though much less perfectly—by fixing one coupling to overhang the shaft, and the opposite one with the shaft projecting. In all cases, the abutting surfaces of the two couplings should bear well in proximity to the bolts. They may stand

very slightly apart near the centre. The bolts are sometimes applied with their heads and nuts projecting, which is very dangerous in work. These are therefore usually recessed, or an overhanging flange provided, as shown in Fig. 131, which gives protection against

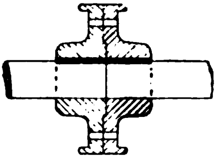


Fig. 131.—Flanged couplings, keyed to shafts.

accidental entanglements. This class of coupling is very efficient, but is more costly than the plain muff coupling, against which some saving is effected by dispensing with the bossing of the shaft-ends. It is very convenient for taking apart and replacing, and in many cases the system is adopted with this especial object. Some engineers force these couplings on the shaft, which in combination with a key gives absolute security. But serious trouble and loss of time in making changes of pulleys are thus involved, unless—as then becomes best—split pulleys are adopted throughout the work.

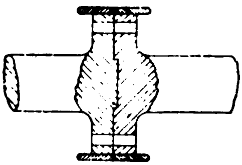


Fig. 132.—Flanged couplings, forged upon shafts.

Face-plate couplings solid with shafts.—Face-plate couplings forged solid with the shaft are sometimes used in good work. One is spigoted into the other, as shown in Fig. 132. Bolts are fitted, as in the last case.

Wheels or pulleys which may be applied to such shafts must be made in halves, or made with holes sufficiently large to pass over the flanges, and fitted on split bosses attached to the shaft by bolting or keying. The smithing work upon such shafts is costly, and they are seldom adopted except for shafts of such large sizes as would be forged in any case. They are especially suited for the propeller-shafts of ships and for underground work, where the confined space available for driving keys is apt to lead to slackness and insecurity. Solid couplings of this kind are rather narrower for passing through confined spaces than separate couplings keyed on the shaft. Obviously solid couplings need only be employed upon such parts of a long shaft as requires them, and in any case the two end lengths are free to receive ordinary unsplit wheels or pulleys. Protecting flanges which are easily provided on cast-iron face-plate couplings are very costly and troublesome to provide solid on the forged couplings. If the situation is such as to render them necessary, they should therefore take the form of separate rings secured by shrinking or by recess-headed screws.

Bolts necessary for face-plate couplings. The number of bolts used in each pair of couplings may be varied to suit different circumstances. When, however, there is a free choice one bolt may be adopted for each two inches in the diameter of the shaft, with the addition of four bolts in each case. The diameter of iron bolts should be sufficient to allow ·75 ton per square inch shearing stress on the bolts for a 4-inch shaft, and 1·00 ton on those of a shaft 10 inches or more in diameter, or stresses one-fifth greater upon steel bolts. The sectional area of the bolts for this purpose may be taken along the surface of contact between the two couplings, *i.e.* upon the full area of the holes into

which the bolts are driven. The amount of shearing force to be resisted by the total number of bolts in tons =

$$\frac{\text{Indicated horse-power} \times 28{\cdot}14}{\text{Revolutions per minute} \times \text{radius to centres of bolts in inches}}.$$

For this purpose, the maximum horse-power should be taken, even if only of momentary application, and allowance should be made on account of any shocks which may be likely to arise in work. The proportions and numbers of bolts adopted may be varied to suit exceptional cases, but in all cases the total shearing strength should be suited to the radius at which the bolts are placed. In no case should bolts be adopted less than three-quarters of an inch in diameter. Bolts proportioned as above will in all ordinary cases provide ample bearing area in the bored holes to support the pressure due to the work transmitted. But for this condition to be efficiently fulfilled, the bolts must fit the holes with such tightness as to require driving by means of a heavy hammer or an hydraulic press. This is important in all cases, but especially so in shafting which is driven in alternate directions when fully loaded. If in the latter case the bolts are slack, slipping will occur at each reversal, unless the strength of bolts is very much greater than required for their legitimate work.

Compound couplings.—Many patterns of compound couplings are made by different firms, of which, perhaps that made by Messrs. Sellars of Philadelphia is the most widely known, and may be taken as representative of the class. In connection with these, the shafts are made quite plain, and free from key grooves. Planished or plain turned shafting may therefore be used with no other preparation than cutting off to length. The outer part of the coupling is a shell or ring corresponding to

the plain muff coupling, but rather greater in length and less in diameter. The shell is bored conically from each end to receive two conical bushes, partially split for flexibility, which are placed in position between the shell and the respective shaft-ends, and forcibly drawn, by means of longitudinal bolts, into driving contact with the shaft-ends. The angle of the cones is such that on slackening the bolts the cones are loosened, or may be started with ease, the whole of the surfaces having been lightly smeared with oil before originally putting together, so as to prevent seizing. The internal diameters of the conical bushes will readily follow such slight differences as may accidentally arise in hot- or cold-rolled shafting, and different bushes may be used to suit different nominal diameters of shafting, with the utmost convenience. These couplings are about as costly as those of the face-plate type, varying somewhat according to the proportions of each, as adopted by the particular maker considered. They are in every way convenient to manipulate, neat in appearance, and free from serious liability to accident. But if not well tightened, they are liable to an occasional slip, from which defect all those secured by a sunk key are free. Many compound couplings are open to considerable improvement by means of increased strength of parts.

Positions of couplings.—All couplings are placed as close to the bearings of the shaft as possible, and on that side of the bearing which is furthest from the driving point or source of power. The chief reason for this is that if any part of a shaft requires to be stopped, the proper coupling may be taken off, and the shaft will still be supported throughout the running part. A second reason is that the shaft is generally secured against longitudinal movement at the driving end. The facility for temporary longitudinal movement thus

afforded is a great convenience, often obviating the hoisting of the shaft—or, at all events, the chief part of it—when disconnected for making changes.

Uniformity or variation in diameter of shafting.—In a large works subject to frequent re-arrangements of machines, it is almost necessary, and certainly very convenient, to have all the line-shafting of one size, or at most of two sizes. In some cases, also, the majority of the driving-pulleys are of equal diameter, and of only one or two widths. When, however, both shafts and pulleys vary largely, the confusion becomes almost intolerable, though subject to considerable relief by means of a well-arranged system of conical bushes for fixing the pulleys upon the shafts. Where the machinery arrangement is of a more permanent character, the shafting may be reduced in diameter at each coupling, or each second or third coupling, so as to effect a considerable saving in original cost and in frictional resistance to driving. If in each length of shafting, an equal amount of power is distributed, and the diameters throughout worked in strict accordance to power, the end length will probably prove too light for stiffness, and the centre length greater than the mean diameter. But by slightly increasing the diameter of the end length, the whole series may be varied by uniform steps or alternate ones.

Speed of shafting.—The most advantageous speeds for shafting are, as a rule, from 120 to 240 revolutions per minute. In some cases a higher speed is adopted for a shaft which drives a number of quarter-twisted belts, as by this means the diameter of pulley and obliquity of belt may be reduced. Sometimes a high speed is adopted on account of the smaller diameter of shafting which becomes possible, whereby a reduction in original cost is secured. But this measure is very apt to lead

to undue sacrifice of transverse or torsional rigidity of shafting, and when the shafts are accurately proportioned in diameter and speed, the loss by friction under ordinary conditions of lubrication remains unaffected. In exceptional cases, to suit very high speeds of machinery, either throughout a works or in any particular department, a higher speed of shafting may be adopted with advantage, for the purpose of improving the transmission by avoiding or reducing the disparity between the sizes of pulleys connected by each belt.

Collared and thrust bearings.—Each length of shafting requires to be restrained against end movement within certain limits, which vary with the conditions. This is usually effected by means of collars welded on the shaft, on each side of one bearing in each connected length of shafting. Collared bearings, being generally placed in close proximity to wheels or main pulleys, should be made of full diameter, or a little over, so as to allow good bearing surface. As in all other cases, changes in diameter should be made in such a manner as to maintain approximately uniform elasticity throughout the length of the shaft. For a very heavy shaft two bearings are required close to the driving pinion or pulley. Both these are usually collared, but this is seldom necessary in connection with rope or belt gearing, and in a second motion shaft for toothed gearing collars are generally more useful near to bevel-wheels. Longitudinal thrust due to bevel-wheels or other causes, when acting towards an end of the shaft not more than 30 feet distant, may be met by the provision of a disc-plate applied to the end of the shaft. In such a plate the centre part of the surface should be cut away to a diameter about one-third that of the outer boundary of the bearing. This is partly to give access for lubrication, but chiefly to promote uniformity of bearing.

When a solid plate is used, the edges wear more rapidly than the centre part, which leads to seizing at the centre, and in many cases to great trouble by heating. Under other conditions, one or more wide collars may be provided to receive the thrust upon solid cast-iron blocks—with or without brass facings—each regulated by screws fitted with nuts on front and back. As to the areas of such plates and collars, reference may be made to the chapter on friction and lubrication. Solid inadjustable blocks, to fit a number of narrow collars, according to former practice in propeller shafts, should in all cases be avoided. In some cases the separate cast-iron blocks for collars must be made in halves, very accurately and firmly secured together. They should, however, always be made solid where possible, even at the cost of a considerable amount of trouble in design and construction.

Pressure upon bearings due to work transmitted.—In the transmission of work, pressure is imposed upon the teeth of one wheel by those of the other wheel. If the teeth could make practical contact only along the line of contact of the rolling cylinders or cones, and if the surfaces of the teeth could be so formed that the surface of each tooth when in contact with the opposite one would be radial, the only pressure between the two would be that due to the power transmitted from one wheel to the other, expressed along the line of motion. In practice, however, this is impossible for many reasons, and a very considerable pressure is developed in a direction tending to separate the centres of the wheels. In spur-wheels of all proportions ordinarily adopted, the secondary force—as it may be termed—may be neglected. But in connection with bevel-wheels, the thrust in a direction normal to the cone of revolution gives rise to longitudinal pressure acting along each

shaft, which must be resisted either by collared bearings, thrust bearings, footsteps, or by disposing different pairs of bevel-wheels so that one-half of the pressure is exerted in each direction.

The useful force applied to the teeth of a wheel, and which acts along the line of motion at the point of contact, also gives rise to a force equal in amount, and opposite in direction, upon the bearings supporting the shaft. In horizontal shafts whose centres are at the same level, and which are connected by wheels, the pressure between the teeth due to the work transmitted will cause an increase of equal amount in the vertical pressure due to weight upon one bearing, and an equal decrease upon the bearing of the other shaft, one thus sustaining more and the other less pressure upon the bearings than when at rest. If two parallel and horizontal shafts, of which one lies vertically over the other, are connected by spur-wheels, the pressure upon each bearing due to the useful pressure upon the teeth becomes expressed in a horizontal direction, parallel but opposite in the two cases. In the latter case such pressure will, in combination with the pressure imposed upon the bearing by reason of weight of parts, produce an actual working pressure in an oblique direction. An infinity of intermediate conditions may also arise. Problems of this class may be easily solved by the use of the parallelogram of forces, those affecting spur-wheels by reference to two dimensions only, and those applying to bevel-wheels by reference to three dimensions.

Longitudinal freedom in bearings.—The bearings of a shaft are found to remain smooth in work for a much greater length of time if the shaft is allowed a little freedom in a longitudinal direction. The amount of this which should be allowed is very much greater in some cases than others. When the shaft carries bevel-

wheels up to 3 inches pitch, about one thirty-second of an inch should be allowed, and about a sixteenth when the wheels are of 6 inches pitch. If all connections are effected by means of unshrouded spur-wheels, the allowance may be from one-eighth to one-quarter of an inch. If all connections are made by means of belts or ropes, from one-quarter to three-quarters of an inch may be allowed. Brass expands by heat to a greater extent than iron, and for this reason some allowance should be made, even if the first reason should not be considered sufficient. If both the brass and the shaft are uniformly heated to a temperature 360° F. above the original one, the brass will be found to have expanded $\frac{1}{250}$th part of its length, and the iron $\frac{1}{400}$th, the difference being $\frac{3}{2000}$. But in many cases the temperature of the brass rises much more rapidly than that of the shaft, and usually an increase of 360° F. in the brass, and one of 180° in the shaft, should be provided against, or an allowance made of $\frac{1}{360}$th part of the length of bearing.

Positions of collared bearings, and use of expansion couplings.—A shaft which drives, or is driven by one pair of bevel-wheels, should be provided with a collared bearing as close to these as practicable. When several pairs of bevel-wheels are carried by the shaft, the collars should occupy an intermediate position. Wheels with helical teeth admit of less freedom than those with plain teeth. Under ordinary conditions, when several pairs of large bevel-wheels are carried by a shaft at points more than 40 feet apart, the expansion and contraction of the shaft by reason of variations in atmospheric temperature assume considerable importance, and an expansion joint becomes desirable, while each section of the shaft is provided with a collared bearing. The expansion joint may be placed in any position intermediate between

the two wheels, regard being paid to the necessity for steadiness in each wheel. The best position is usually about half-way between the two wheels. Ordinary claw couplings are sometimes used as expansion couplings, but a better and more usual form is a solid cylindrical coupling, into the bore of which the end of each shaft reaches to the centre of its length. The end of one shaft is securely keyed, and the other is driven by two sliding keys, fixed in grooves on the opposite sides of the shaft. The shaft and keys are made to fit accurately, but easily, allowing expansion and contraction to take place freely. Obviously, expansion couplings are not necessary in connection with a shaft carried by a continuous structure of iron, as the two expand together.

Fitting and keying of wheels.—Large wheels or pulleys are usually fixed to a shaft by four keys, as described in connection with fly-wheels, the shaft being bossed to a larger diameter, and the hole in the wheel made to a still larger diameter. Wheels of moderate sizes are usually bored to fit bosses provided on the shaft, and secured by one key, which fits a flat on the shaft and a key-way in the wheel. In either case preparation must be made to slip the driven wheel out of gear, so as to stop the shaft with comparatively little trouble. When many wheels are required on the same line of shaft, they are of comparatively small size, and are sometimes fixed on a plain parallel shaft by conical cast-iron keys. These are also very largely used for small unsplit pulleys, and are the only means for convenient and efficient use upon the main body of shafts, which are bossed to receive either wheels or couplings. A conical hole is bored in the boss of each pulley, whose smallest diameter is sufficiently large to allow the pulley to pass the bosses on the shaft. This hole is fitted with a bush

which is bored to gauge fitting the shaft, turned on the outside to fit the pulley, and split longitudinally in three equidistant places. The three pieces so produced are driven into the pulley boss, and held by friction alone. They are easily fixed or removed, and if the pulley should require to be transferred to a shaft of different diameter, new keys can be easily made. A pulley secured in this way is shown in Fig. 133, but the amount of taper being on each side only 1 in 50, or 1 in 60, is too small to show in the figure. Conical keys hold with great security on shafts of all sizes, but require to be well driven when a large pulley is fixed

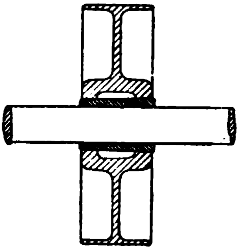


Fig. 133.—Pulley secured by conical keys.

on a small shaft, as then the radius at which friction acts is comparatively small. In such cases also the boss must be sufficiently strong to withstand the driving.

Precautions against injury by shock.—Shafting and wheels exposed to great shock in work require to be provided with some means whereby they may yield, and so avert damage, which in some cases cannot be prevented by any practicable accession of strength. The simplest means whereby this can be achieved is by the adoption of frictional transmission, especially by belts. Rope transmission and smooth wheels which drive each other by frictional contact also meet the case in many instances. In some cases it is, however, most

desirable that toothed-wheel connections be adopted throughout, when one wheel, generally of large diameter, is selected for fitting as a slip or surging-wheel. This is provided with a centre of as large diameter as possible, which is turned smooth over the circumference. The toothed rim of the wheel is provided with an inner rim fitting the centre, so as to revolve loosely upon the latter. The inner rim is fitted with a number of cast-iron or brass blocks, bearing accurately upon the turned surface of the centre-piece, and tightened by strong set-screws. The sliding force is greater than the pressure upon the teeth of the wheel in proportion to the reduced radius at which sliding may take place. With cast-iron upon cast-iron, the co-efficient of friction may be taken ·10, so that the combined radial pressure upon the whole of the blocks is ten times the sliding force along the surface. The total surface provided in the blocks should be such as to allow a pressure of 100 pounds per square inch. Several firms now manufacture clutches, wheels, and pulleys, with arrangements for allowing surging to take place under control of a spring. In some cases the spring takes up the work, and in so doing becomes coiled or compressed, the inverse process following in due course. In others slip takes place controlled by the spring, so that it is not followed by any reaction.

Deviation from straight line of shafting.—The line of a shaft is sometimes broken so that the two parts stand at a small angle with each other. A pair of angle wheels is usually adopted at such a junction, and may be considered to be of universal application. Hooke's universal joint is, however, most successful in many cases. Fig. 134 shows such a coupling, which was applied in 1886 by Messrs. Simons of Renfrew on Dredger B.D.I. to transmit 300 to 400 indicated horse-

power at 100 revolutions per minute, the deflection from a straight line being $3\frac{1}{4}°$. A coupling of this kind may be used for angles up to 5° or 10°, according to the degree of uniformity in transmission required. Two couplings similarly disposed will give uniform transmission over an angle of 30°, the trunnion centres upon the two ends of the intermediate length being situated upon the same plane.

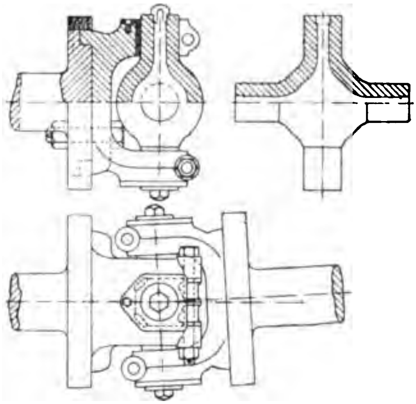


Fig. 134.—Universal joint for shafting.

Strength of shafting against torsion.—The strength of shafting has been discussed in connection with crank-shafts. In all ordinary cases the work is transmitted with practical uniformity, and the diameter in inches which is necessary in any case =

$$\sqrt[3]{\frac{\text{Indicated horse-power} \times 143}{\text{Intensity of stress upon the surface of material in tons per sq. in.} \times \text{number of revolutions per minute.}}}$$

Table XXVI. gives the horse-power of iron shafts per revolution performed in one minute, and in each case the number of teeth in an uncut wheel of cast-iron of equivalent strength. The intensity of shearing or

torsional stress upon the shaft, corresponding to the values given in the table, is 1·76 tons per square inch. Steel shafts may be loaded to 2·20 tons per square inch, to transmit one-fourth more power than given in table. The diameter of a steel shaft loaded with such increased stress will be 7·2 per cent. less than that of an iron one, and its weight will be about 11·5 per cent. less. Shafts thus proportioned possess good torsional stiffness for ordinary work. If, however, torsional stiffness is not of importance, such diameters may be reduced, but under no ordinary circumstances should a shaft of iron be loaded beyond 2·5 tons per square inch, or one of steel beyond 3 tons. A shaft which is steadily loaded attains a constant condition of torsional deflection, and therefore may be loaded to a higher point than one which is liable to fluctuation. For this reason mule-room shafts should be made rather larger than given in the table. Shafts of different diameters, accurately proportioned as to strength, suffer torsional deflection (by angular measure) in inverse ratio to the diameters. But with smaller shafts, driven at a correspondingly increased speed, smaller pulleys are required for the same belt speed, so that the effect upon the linear motion of the belts caused by torsion of shafts is greater with the larger shafts in the proportion of the square of the diameter of the shaft.

Strength of shafting against bending.—Line-shafts are not usually exposed to transverse loading to such an extent as to approach the safe strength of the shaft. In all ordinary cases, shafts of diameters in accordance with Table XXVI. are found to be free from objectionable transverse deflection. In exceptional cases, the deflection may be estimated according to the rule already given in the chapter on crank-shafts, and if found to be excessive, the diameter of the shaft should be increased.

If the loading is constant, the bearings may be canted to suit a moderate deflection; but if variable and considerable in amount, swivel or self-adjusting bearings should be adopted, as solid bearings will quickly wear away, by reason of the constant changes in the distribution of pressure upon the bearing surface. When a main driving or driven wheel or pulley of any kind is mounted on the over-hanging end of a shaft, such shaft should be provided with a bearing as close to the wheel as possible. The strength of such a bearing should be twice as great as that of an accurately-proportioned shaft, or its diameter one-quarter greater, and in some cases a further increase will be necessary on account of the amount of pressure upon the bearing.

Upright shafts.—Upright shafts, as to strength, are subject to the same rules as horizontal ones. The whole weight of the shaft and the wheels, etc., carried by it, is generally supported on a footstep placed under the lower end of the shaft. The weight is generally more than sufficient to prevent the shaft from rising in work, so that no collars are required for the purpose. The extreme lower end of a heavy shaft is made of steel to give a tough and smooth surface, even when the rest of the shaft is of iron. This runs in a gun-metal bush, which is sometimes whole and in other cases split. A plate or disc of tough gun-metal is placed beneath the shaft to support the weight. This plate is often provided with a wrought-iron hoop, shrunk on, to prevent it from splitting under pressure. As in the case of thrust bearings, at least one-third of the diameter in the centre should be cut away, and adequate preparation made for abundant supply of oil. A cap is provided in front, which forms an oil-tight joint with the block, preventing the escape of oil, and, on its removal, allowing free access for inspection of the

shaft and the brasses. The brasses are released for withdrawal by slightly elevating the shaft; and lifting-screws for effecting this are provided in the first instance.

Pressure upon foot of upright shaft.—The footstep of an upright shaft ought not to be loaded to more than 200 pounds per square inch. In many cases, however, this condition cannot be observed, and loads of 300 to 400 pounds per inch are imposed, when the work can only be kept going under ordinary conditions by the exercise of the most careful attention. In such cases the footstep may be closed by the application of a stuffing-box, and oil forced in under hydraulic pressure sufficient to reduce the load upon the metal to 200 pounds per square inch. The footstep may also be arranged so that oil may be forced into it under pressure, when the load may reach 1 ton per square inch in safety, so long as a copious supply of oil is provided, without the necessity for a stuffing-box to surround the shaft-end.

Bearings of second-motion shaft.—The bearings of second-motion shafts are practically reduced copies of the flat and angular pedestals used for crank-shafts. When toothed gearing is adopted, the second-motion shaft is usually run at about 120 revolutions per minute. It is generally placed in the same horizontal plane as the chief lines of shafts, with a view to direct driving by simple bevel-wheels. The engine is often erected with its crank-shaft on the same level, but more frequently 6 or even 10 feet lower. The shafts for the upper rooms are arranged in the same vertical plane, one upright shaft placed directly opposite the series suffices to drive the whole, and only one pair of bevel-wheels is required for each storey. In most cases, the pedestals near to the main pinion are of the angular type, and all the others flat ones, with or without the addition of special thrust bearings.

Bearings in connection with belt or rope gearing.—In deciding the proportions of bearings for use in installations of belt or rope gearing, full account should be taken of the tension upon the slack side of the belts or ropes, which question is treated elsewhere. Separate second-motion shafts are generally adopted, crossing a race in such manner that special care is necessary to ensure that no shaft shall be exposed to an intensity of bending stress exceeding 3·2 tons for iron, or 4·0 tons per square inch upon steel. The torsional strength, the amount of deflection allowed, and the permissible intensity of pressure imposed upon the bearings, may be treated as in a crank-shaft. The supporting bearings ought to be as near to the pulleys as possible, to obviate the necessity for large shafts. In some cases an arrangement is adopted which is practically equivalent to the self-adjusting bearing subsequently referred to. The pedestals are held to a fixed horizontal plate, by one holding-bolt to each end of each pedestal. Passing from one of these bolt-holes to the other a ridge is provided on the fixed plate, or on the base of the pedestal, which ridge projects from the general plane of the face of the casting; or a loose strip of some material is inserted in the same position, which lies across the shaft in plan. The nuts of the holding-bolts are gently screwed down fairly tightly and locked. The pedestal will then follow the shaft when the bearing part of the latter is tilted by deflection, the whole length of the bearing is uniformly loaded, and overheating is prevented, or stopped if not too far advanced. This plan is obviously most applicable to cases in which the ropes or belts approach to the vertical position.

Line-shaft bearings and pedestals.—Line-shafting is usually supported in bearings placed about 10 or 11 feet

from centre to centre. This distance is usually decided by considerations in connection with the building or the machinery to be driven, rather than with the shafting itself. The conditions are, however, seldom such as to lead to the imposition of excessive bending force upon the shafting, and generally the work is found satisfactory. But if, to give more space around the machines, or for any other reason, the distances of columns and beams—which are usually utilized to carry the shafting—should be materially increased, it will be wise to increase the number of bearings. The bearings are usually fixed, that is, without facility for self-adjustment. The length of bearing surface is equal to twice the diameter. Top and bottom brasses are provided, or bottom brass and cast-iron cap to fit the shaft and bored with the bottom brass. The first is usually adopted when the shaft is subject to an upward pull of considerable magnitude. The second may be applied when a moderate pull against the cap is anticipated. In cases where the weight is constantly upon the bottom brass, the upper brass may be replaced by a hollow or "shell" cap, which may be charged with tallow for use in case of accidental heating of the bearing. The brasses for bearings in proximity to main wheels and pulleys are turned, and carefully fitted into bored pedestals, as described in the chapter on cranks and main bearings. The brasses for the normal support of line-shafting are generally cast very near to size outside, so that a little filing will suffice to fit them in the pedestals, after which they are bored out. If, however, at any one point a large amount of power is taken off a line-shaft, the next adjoining bearing should be fitted as a main bearing. All pedestals are fitted with removable caps, so that the shaft may at any time be easily lifted out without disturbing any pulley, and so as to adjust for

wear, though this should be very little in shafting well-proportioned and fitted. The majority of pedestals are required for attachment to the brick- or ironwork of a building, in some cases to a vertical surface, but more frequently to a horizontal surface. A cast-iron beam or column may be easily prepared to receive a pedestal to be held by bolts at each end, arranged for allowing a little adjustment to meet original inaccuracy in the building, or on account of subsequent settlement. Snugs are also provided at one or both ends, and keys of beech or iron are fitted against the pedestal, so that slipping at the bolted surface is impossible. In important bearings the pedestal, and the bed upon which it rests, are usually planed and bolted "metal to metal." In other cases a thickness of wood is inserted. The latter is generally considered as obsolete practice, but it is sometimes of great value in providing a little elasticity, and promoting smooth working. The wood should, however, be limited in thickness to, say, $\frac{5}{16}$ inch in pine, $\frac{1}{2}$ inch in birch, and $\frac{3}{4}$ inch in hard oak; but less thickness will generally be quite sufficient; planing of surfaces is still of importance, though not absolutely necessary. Brickwork, and in many cases wrought-iron constructions, do not lend themselves to accuracy and strength of direct attachment of pedestals. Consequently plates, brackets, and wall-boxes are adopted. These are built in the building, or immovably attached to the surface, and are described in another chapter. The pedestal may be attached to those by means of ordinary bolts passed through the whole, or by means of T-headed bolts, when access for passing through a bolt is impossible, or when it is desired to prevent all possibility of leakage of oil. In some cases, where space is limited, one or both ends of a flat pedestal are cut off, and the cap-bolts extended downwards to act as holding-bolts, these

then become collar-bolts with a nut at each end, and require corresponding access for screwing up.

Side pedestals are bolted to columns specially prepared to receive them, or to some other surface approximately vertical. They are fitted with double or single brasses or with shell caps, precisely in the same manner as flat pedestals. They are almost always bedded against wood, and usually are keyed against a lower snug, but rarely against an upper one.

Self-adjusting bearings.—Bearings which possess a facility for self-adjustment with respect to the direction of the axis of the revolving shaft have been adopted in English practice in many isolated instances, but not generally. They are largely adopted in the United States, one of the chief reasons being that the buildings, being lighter, and containing more wood in their structure, are not so stable as those in England. Flexibility of building construction, however, sometimes occurs in England to an objectionable extent, and bearings of this class might be more extensively adopted with advantage. Their adoption naturally leads to reduced diameters of shafting, which is in some cases an advantage, and in others a disadvantage. Bearings of this class are generally made with convenient means of vertical adjustment, by which time is saved in erection, or in subsequent adjustment on account of vertical subsidence in the building. Side adjustment is secured in precisely the same way as in bearings of the ordinary class. The swivelling action admits of the use of bearings of double the usual length, by which means the pressure per square inch of bearing surface is reduced, and the use of cast-iron for the bearing surface is rendered practicable. As a rule, this pressure is not allowed to exceed 56 pounds per square inch.

Access and light.—It is very important that each

bearing should be completely accessible when at work, and, if possible, in full daylight.

Counter-shafts.—In textile works and corn mills, each machine must be either in motion at one uniform speed or entirely stopped. Consequently it is sufficient to provide a pair of fast-and-loose pulleys on the machine itself, and on the line-shaft a pulley of a suitable diameter, and of width sufficient to accommodate the belt when on either pulley of the machine. But for machine tools of all kinds, and for other classes of machinery, convenient means for varying the speed must be provided. This is found to be most efficiently secured by the use of a counter-shaft with a cone pulley upon it, to match one upon the machine. Counter-shafts are only required to drive one machine, and for the sake of uniformity and interchangeability are often made of much greater strength than necessary for the work done. On the whole, this is a wise measure to adopt, though it causes some loss of power. In some cases where a large number of counter-shafts are employed, it is found that one-half or more of the total indicated horse-power of the engine is required to drive the shafting and counter-shafting or loose pulleys when all machines are stopped. In many cases, more power would be saved by the application of self-adjusting bearings to the counter-shafts than to the line-shafting. In some cases, counter-shafts are adopted for convenience in increasing the speed from that of the line-shaft to that of a high-speed machine.

CHAPTER XLII.

WALL-BOXES, BRACKETS, AND PLATES.

Wall-openings secured by boxes.—Openings are often required to be made in walls of brick or stone, for the passage of shafts, belts, ropes, pump-rods, and other details. These openings should be made as small as possible, with a view to the maintenance of strength and rigidity, and to prevent or obstruct the passage of sound and fire. Wall-boxes of the simplest type are used to make good such openings, but not for the support of the object passing through. Such boxes are made of cast-iron, and flanges should be provided along the inner edges with bolt-holes for the attachment of strong wrought-iron plates to act as a double fire-proof barrier. The opening made for the passage of a shaft or rod may fit very closely, so that the efficiency of the barrier is not affected; in some cases a small box of circular form cast from a heavy pulley pattern is good. But with a belt or rope more clearance is required, which necessitates a larger box. The amount of clearance in the slits should be minimized by the provision of guide-pulleys at a short distance from the slit. A guide-pulley placed inside the box is often effective for the slits of both plates.

Wall-boxes to carry shafts.—A wall-box may often be conveniently modified to provide support for a shaft. A bridge to receive a flat pedestal is thrown across in the form of a flat plate stiffened from below and provided with slots for holding-bolts and snugs for securing the pedestal from slipping. In all cases tongues or other means should be provided, with a view to completely prevent oil, grease, or dirt of any kind from spreading to surrounding surfaces or objects.

Construction of wall-boxes.—Small boxes are cast in one piece, but large boxes made in this way are liable to crack, often without apparent cause. Solid boxes are sometimes made with an arched top for greater strength to support top weight. No direct advantage can be derived from such an arrangement unless very great care is taken in packing the box all round, and especially at the top of each side, so as to prevent the spreading of the sides by reason of the thrust from the arched top. An indirect advantage thus secured consists in the use of such an arched box as a centre upon which to build a brick arch. Large boxes are always made in the form of separate plates bolted together in position. These must be arranged so that the pressure imposed by the surrounding brickwork or masonry shall be resisted by abutting surfaces of cast-iron, the bolts being used to prevent the plates slipping apart. In new work, ordinary through-bolts with square shoulders fitting in oblong crossed holes are adopted. A wall-box is, however, often inserted when alterations are made in existing work. In such cases a solid box may be adopted, to a larger size than in new work, as plates are difficult to fix soundly and satisfactorily. The best connection is then by means of set-screws fitted in holes tapped in the plates. In all cases greater security is obtained by the use of outward flanges to

fit against the solid wall on all sides. It is not practicable, however, to provide these on all sides, front and back, of a box used to place in an existing wall. In some cases the bridges or beds for one or more pedestals may be built up from the bottom plate. In other cases bearers are fitted across the boxes after erection, and to suit lines which cannot be absolutely determined until the building is well advanced. The wall face flanges may also be enlarged or modified to receive plain pedestals or brackets to carry the same. The positions of large wheels are designed, when possible, so that the bearing may be supported by wall-boxes, for the sake of the stability imparted by the weight of the wall, and the reduction in vibration which is secured. In best arrangements for the support of a main upright shaft, a large box is provided for the wheels at the bottom of the shaft, the bottom plate of which is continued upon a brickwork foundation, carried up from the ground and finished with stone cap-blocks. Holes are built or drilled through the foundation, reaching down to hand holes near the ground level, or, if possible, not less than eight feet below the top. The foundation is bonded into, and carried up with, the main wall. A brickwork tower surrounds the foundation, and is carried to the top of the building, or as far as the shaft, and securely covered over. This arrangement, shown in Fig. 138, provides convenient and secure access to the wheels, shafts, and bearings at any time, gives protection against the spread of fire, and ensures cleanliness. In some cases a walled passage is provided, to similarly isolate the second-motion shaft throughout its length.

Fixing of wall-boxes.—Wall boxes are most conveniently and securely fixed in a wall when built in as the wall is erected. A large ashlar stone is placed beneath each box, the top bed is dressed off, and the

bottom plate of the wall-box is evenly bedded in cement. The sides and top are then erected upon the bottom plate, and the brickwork built closely against the sides until level with the top, when a second large stone is placed over the box, resting on the wall at each end. The stone should not press very heavily upon the centre of the top plate, or the latter will probably be broken. But the space between the box and the lintel stone should be well grouted, after allowing to settle. When moderate-sized boxes are inserted in existing work in very sound condition, the stones may be dispensed with, so as to reduce the amount of material cut away.

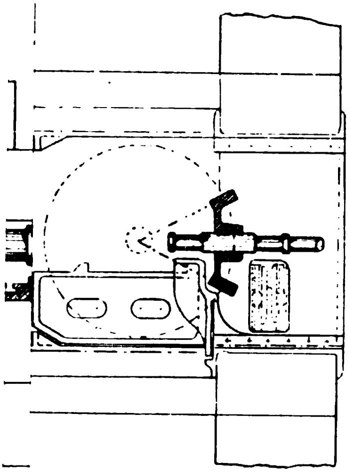
Face boxes and plates.—Wall-boxes which do not give an opening through the wall are adopted for carrying brackets for the support of a shaft running parallel to, and at a short distance from, a wall. Such boxes give great security, combined with the means of original and subsequent adjustment to the brackets, in a degree quite unattainable when the brackets are bolted directly to the wall. Fig. 137 shows such an arrangement. A plate equivalent to the front plate, provided with wall-lugs on the back, and well bolted by bolts through the wall, or roughed bolts secured by cement may also be used. Bolt-plates and similar details bedded against brickwork should be arranged with ample surface, so that the pressure per square inch upon brickwork of moderate quality shall not exceed 150 pounds per square inch, or, upon brickwork of excellent quality, 200 pounds per square inch in absolute contact.

Brackets.—A bracket, or knee bracket, is used to support an ordinary pedestal in a position near to, but not within, a wall. As a rule, T-slots should be used to receive the pedestal holding-bolts, so that no drip of oil may follow the holes which would be required

for through-bolts. This arrangement is, however, not popular, owing to the slightly greater difficulty, and additional hoisting of shafts involved in moving pedestals.

Support of counter-shafts.—Isolated counter-shafts may usually be carried either by simple brackets against a wall, or by hanging brackets from an upper floor. In the former case, the belts from the line-shafts will usually run vertically, and in the latter case often horizontally. When, however, the whole of the floor space is closely filled with machines of small or moderate size, the number of counter-shafts becomes too great to be dealt with by ordinary means. Bearers or stringers may be placed along the whole width of the room, at right angles to the direction of the line-shafts, at some convenient distance below them. These will carry any number of counter-shafts in the most efficient manner, by means of flat pedestals, adjustable to any position by means of T-slots in the bearers. Strap-shifting gear is easily dealt with, and planks laid across the bearers will at any time give an excellent platform for access to either line-shaft or counter-shafts. In a side-lighted room, a certain amount of height is necessary, in order to avoid darkening the room by this means. The use of cranes for serving the machines is also very much restricted, but in such cases cranes are usually of minor consequence. In all other respects the system is unrivalled. Double ranks of bearers may be adopted if desired, or the system may be confined to a few bays. Self-adjusting bearings may also be substituted for the ordinary pedestals.

General conditions.—In designing wall-boxes and brackets, attention should be given to stability, general accessibility to all parts for oiling, cleaning, adjustment, and repairs, each driven wheel should be easily



(b)

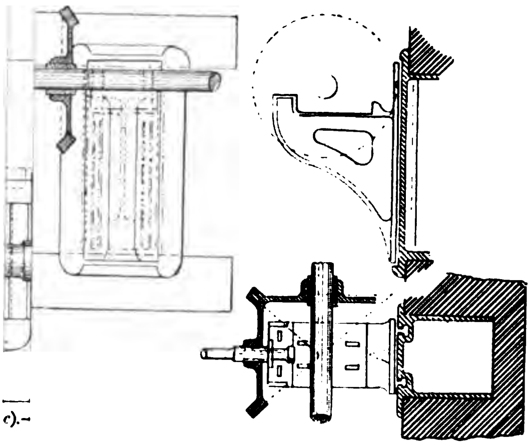
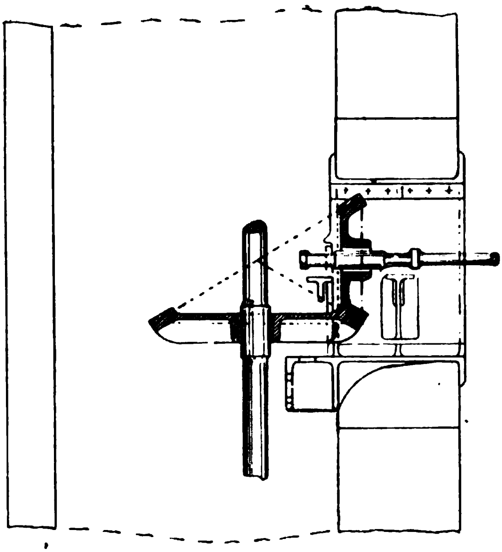


Fig. 137.—Bracket carried by box built in wall.



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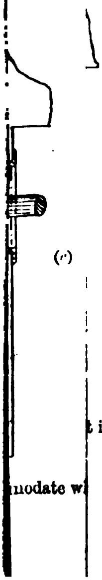


(c)

(c)

t is driven by upright shaft.

nodate w

slipped out of gear, the whole should be free from risk of fire communication, and grease should be prevented from appearing on the face of any wall, ceiling, or floor. Wheels are generally covered by cases of galvanized iron or other metal. Hanging or sliding fire-proof doors should be self-closing, and never allowed to be blocked back when it can be avoided. In exceptional cases, such doors may be held open by combustible cords attached to fusible metal fastenings to impart double security. All these conditions can be fulfilled without proceeding to extravagance in the original cost of the work, and when once secured they will give pleasure and economy in use.

Figs. 135 to 139 give examples of wall-boxes and fittings suitable for use in connection with toothed gearing. Similar boxes, smaller and more lightly constructed, are suitable for use in connection with belt and rope gearing.

CHAPTER XLIII.

BALANCING OF MACHINERY.

Vibration due to centrifugal force.—Shafts, pulleys, wheels, cranks, and other revolving parts of machinery which transmit uniform work, should run with uniformity and silence. It often happens, however, that an oscillation, knock, or tremor occurs, once or twice in each revolution, by reason of the weight of one side exceeding that of the other, and by reason of the centrifugal force due to the unbalanced weight. An object which revolves in a horizontal plane round a vertical centre tends to fly away or to increase the distance from the centre. This tendency must be resisted by a force acting radially, equal to the centrifugal force, and which in pounds =

$$\frac{\text{Weight of revolving object} \times \text{radius of revolution} \times (\text{number of revolutions})^2}{3000}.$$

Weight in pounds.
Radius in feet measured to centre of gravity of revolving object.
Number of revolutions per minute.

The amount of centrifugal force is precisely the same if revolution occurs in a vertical plane, and according to the direction of motion at the moment, it may act with, against, or independently of, the force of gravity

If a revolving object—such as a crank—is in a perfectly balanced position, and an additional weight—such as a crank-pin—is applied to one side, the balance is destroyed. The measure of defective balance is precisely the same, whether calculated upon the total weight, concentrated at the centre of gravity of the whole, or upon the amount of secondary weight, measured to its centre of gravity, and the amount of centrifugal force will be found to be the same in either case. A moderate amount of centrifugal force thus arising is not objectionable, provided that the bearings are well fitted. If, however, the bearings are loosely fitted, a small amount of centrifugal force will cause the shaft to oscillate from side to side at the bottom of the bearing, without rising up either side to any appreciable extent. As the centrifugal force increases, the irregularity of motion increases, until a condition is reached in which radius $\times$ (revols.)^2 = 3000, when the centrifugal force equals the weight. At this point, unless the revolving object is held by bolts, springs, etc., the whole becomes unstable; and when this is exceeded, the whole will rise and fall, and alternately work to each side with each revolution, as far as the condition of fitting of the bearings will allow. In an extreme case it is impossible to fit the work, so as to avoid vibration, which proceeds in defiance of any practicable strength of shaft or framing. This trouble becomes exceedingly acute in details working at a high speed, even if the balance is apparently only slightly incorrect. A detail, of which the centre of gravity is incorrect by one-quarter of an inch will become unstable at a speed of 400 revolutions per minute, and one working at 800 revolutions will be unstable if the centre of gravity of the whole is only one-sixteenth part of an inch from the centre of revolution. In connection

with light machinery running at high speeds, absolute balancing is essential to success; it is, however, of importance in all cases, long before the acute stage is reached.

Balancing of weights in same plane of revolution.—Every disturbing weight should be balanced in its own plane of revolution. In some cases this is impossible, when the closest possible approach to such a condition should be secured, preferably by the use of two counter-balance weights, whose united centre of gravity occupies the position inaccessible to the material weight; but if this cannot be done, one weight should be placed in a position nearest to the correct one. A case of this kind occurs in small high-speed engines, in which the weight of the crank-pin and connecting-rod end can only be satisfactorily balanced in the adoption of a double-throw crank. A precisely opposite course has been often adopted in single-crank engines, and the weight of the crank balanced by the addition of a counterweight in the fly-wheel. This might be supposed to be efficient, as it facilitates turning the engine by hand, but it clearly increases the irregularity of running. In such a case centrifugal force is developed at two points, so as to tend to pull the shaft in opposite directions at different parts, so that one side of one bearing receives more pressure than the opposite side of the same bearing, while another bearing is contrarily affected.

Uniformity of rotation affected by defective balance.—Apart from the variations or pulsations in the pressure on the bearings, due to the unbalanced condition of a revolving object, uniformity of rotation is prejudicially affected. An instance of this occurs in an ordinary unbalanced crank, in which the amount of power alternately absorbed during the ascent and

restored during the descent may equal or exceed 5 per cent. of the total power of the engine, so that the power available during one-half of the revolution is 10 per cent. greater than that available during the other half.

Balancing in detail.—First-class practice in high-speed machinery may be taken as an example to be more largely imitated. Shafting accurately turned all over must of necessity be truly balanced, but any portions non-symmetrical are first balanced in the manner described. In different cases, pulleys, wheels, centrifugal pump or fan discs, or armatures for dynamos or electric motors are employed, each of which is balanced separately and in its separate parts, preferably on its own shaft, but otherwise on a separate mandrill. For the purpose of testing the condition of balance, the shaft or mandrill — with not more than one detail placed upon it of which the condition is doubtful— should be laid upon two smooth and exactly level strips, and rolled into several positions, from which the heaviest side will be determined. According to circumstances, the adjustment will be made by the removal of material, or the addition of plates, etc., well secured. An incidental advantage, of importance in some such cases, is that the several details may be removed and replaced, accidentally or intentionally, in a different order, without affecting the condition of perfect balance.

Rectilinear balancing. — A reciprocating balance is used for the purpose of absorbing work in its ascent and restoring work in its descent, and thereby tending to equalize the work to be performed by an engine, though it may only be possible to apply the work in an intermittent manner. This principle is applied on the largest scale in connection with single-acting pumps.

CHAPTER XLIV.

FRICTION AND LUBRICATION.

Action of lubricant.—Machinery in motion consists almost entirely of details which move in apparent contact with each other. When absolute contact occurs between bare metallic surfaces, abrasion of material is caused, which varies in amount with the nature of the surfaces in contact, with the condition as to smoothness and accuracy of form, and with the intensity of pressure. This abrasion may be reduced in amount, or even altogether prevented by the use of a lubricating medium, which is usually oil or grease of some kind, drawn from most diverse sources. Occasionally tar and other fluid, semi-fluid, and solid substances are used for heavy rough machinery and wheels. In connection with ordinary bearing or sliding surfaces, the office of a lubricant is to float one surface apart from the other, by means of a film spread uniformly over the whole. This uniformity of film can only be fully secured by the application of the very best workmanship in fitting together the surfaces. A shaft with an unbroken surface revolving continuously upon a film of oil will carry the film forward at a reduced speed. If oil is supplied at the entering edge in sufficient quantity,

the separation of contact will be complete. Periodical separations of surfaces give additional facilities for entry of oil, and are almost invariably attended with good results. But if two surfaces are kept in contact by means of heavy, steady pressure, lubrication can only be effected with difficulty. If the continuity of surface is broken, or it is deficient in accuracy, the exit of oil is facilitated and lubrication impaired. A shaft which is accurate in form, but incorrectly balanced, rotated at a high velocity, gives rise to a pulsating action, which may rapidly reduce the amount of oil in the bearing allowing the surfaces to come into contact, and seize or inflict damage upon each other. The same action also arises when corresponding inaccuracies exist in respect to forms other than circular in section. Surfaces which primarily are in perfect geometrical form, may depart seriously from that condition when in work. Such change may arise from the weight of the part itself, or from pressure received from another part, and may be either constant or variable in amount. The crank-shaft of a large engine suffers deflection to a very appreciable amount, by reason of the weight of the fly-wheel. Heavy framing of cast-iron is also subject to considerable deformation. Such deformations must be recognized and in some way compensated, or the parts will fail to work perfectly.

Morin's experiments on friction.—Until a recent period the only important experiments upon frictional resistances of which the results were available to engineers, were those of General Morin, made in 1831—1833. These were mostly made upon plane, level surfaces, well fitted together, though probably not equal to the best practice of the present time. Lubrication was effected by anointing the surfaces before placing in contact. It was found that when the amount of weight

per unit area was insufficient to cause injury to the surfaces in sliding contact, the resistance to motion due to friction could be overcome by a force acting along the surfaces, the amount of which bore a fixed proportion to the total weight upon the sliding surfaces, quite irrespective of the area of surface in contact, and was practically independent of the velocity, though obviously the amount of work required to overcome friction varied with the distance through which the motion of surface was continued. The fixed proportion or fraction thus ascertained was called the "co-efficient of friction." For all metallic surfaces, plane or revolving, lubricated in the manner described, by means of tallow, lard, or olive oil, the co-efficient of friction was found to be from 0·07 to 0·08. When the surfaces were wetted, the co-efficient of friction rose considerably. Soaped plane surfaces of oak gave 0·16. Soaped oak on metal gave 0·20. Leather on metal, oiled, gave 0·15, unctuous 0·23, wetted 0·36, and dry 0·56. Dry metal surfaces gave 0·20. Other experimenters found that copious lubrication of surfaces effected very great reductions in the amount of friction.

Tower's experiments.—About 1884, an important series of experiments were commenced by Mr. B. Tower, on behalf of the Institution of Mechanical Engineers. In the first and second series a horizontal steel cylindrical shaft, 4 inches in diameter, was loaded by means of a gun-metal bearing, 6 inches in length, applied to its upper surface. The experiments were directed to show the influence of weight or pressure incident upon the surface, the velocity of motion, means of lubrication, and the intensity of fluid pressure to which the lubricant was exposed at the several points. In a third series of experiments two pairs of plane annular surfaces, 1 inch in width, and 13 inches in mean diameter,

were employed. A fourth series were made upon a pivot or footstep bearing, 3 inches in diameter.

Friction of cylindrical surfaces.—In the majority of cases the co-efficient of friction for cylindrical surfaces was found to be from 0·001 to 0·002. With an unbroken surface over so much of the bearing as is limited by a chord of 3¼ inches, the pressure per square inch upon the chord rose to 380 pounds before the bearing seized. When the surface was divided by oil grooves, the bearing could not be loaded beyond 200 pounds per square inch without seizing. The actual amount of frictional resistance increased much less rapidly than the pressure, so that the co-efficient of friction is lowest with high pressures, as though a separate portion of the whole friction is due to the viscosity of the oil. The co-efficient was found to rise with increased speed at almost the same rate as it fell with increased load. The oil was applied to the shaft surface by means of a bath against the under side. A thick film of oil was carried up from the bath and became insinuated between the shaft and the brass. When at work the brass must have been wider than an exact fit upon the shaft, to allow for the film of oil, which appears to have usually effected a complete separation of the metallic surfaces, the whole of the friction being due to the motion of the particles of oil upon each other. The co-efficient varied with the class of oil used. Those of high viscosity opposed the greatest resistance to motion, and grease opposed more resistance than oil of any kind. At high temperatures the viscosity falls, and oil becomes more fluid, reducing friction. At a speed of 450 revolutions per minute, with the whole arrangement at a temperature of 60° F., the friction was three times as great as when the temperature was maintained at 120° F. The oil in the

bearing was found to be subject to a fluid pressure, which was greatest in the centre, and less towards each edge of the bearing. The mean fluid pressure over the whole surface, reduced to the chord, almost exactly agreed with the mean pressure due to the weight imposed upon the bearing. The fluid pressure probably opened the brass a little, so as to admit the oil brought up from the bath, and so facilitate lubrication.

Friction of collar and pivot surfaces.—The arrangements of sliding surfaces dealt with in the third and fourth series were such that no point could be sufficiently distant from the edge to ensure a degree of fluid pressure sufficient to maintain an effective separating film of oil between the surfaces. In the third series the co-efficient was found to reach about 0·04, and in the fourth series nearly 0·01. Both of these are much greater than those obtained in the first and second series with the cylindrical bearing. In the third and fourth series the results obtained approached consistently towards those obtained by General Morin, thus pointing to the probability that the solid surfaces were in contact, as opposed to the absolute separation by a fluid film, which occurred in the first and second series.

Experiments made under steady pressure.—In all the experiments a steady pressure, free from shock or variation, was applied to the surfaces. In practical work this condition seldom occurs. The variations in loading, vibration, and end play, to which the majority of bearings or shafts are subject, are advantageous in allowing the frequent admission of oil, and leading to improvement in the condition of the surfaces.

Abnormal friction due to reversal of motion.—In the first series of experiments with the cylindrical shaft, the direction of revolution of the shaft was frequently

reversed, for the original purpose of ensuring greater accuracy in observing the results. It was noticed that for some minutes after such a reversal, the shaft worked very heavily and showed a tendency to seize, but that this gradually vanished, to re-appear on each subsequent reversal in either direction. After a large number of alternations this action was found to cease. The phenomenon appears to depend upon minute fibres distributed over the surface. These are originally formed upon the surfaces by the finishing tools. Those upon the shaft surface are much reduced, and laid smoothly in the process of polishing, and upon both surfaces by the revolution of one upon the other under pressure.

Pressure upon lubricant.—The whole of the experiments show the importance of copious lubrication. They show that when a surface is divided by oil-grooves the weight-supporting power is much reduced. Also that oil in a bearing exists under pressure, and that a greater pressure must be applied to oil before it can enter the bearing except at the receiving edge. The fluid pressure at the edge of the bearing is so low that the oil enters without difficulty, provided that a good supply is presented. When once entered, the film of oil is carried forward in much the same way as an ordinary roller is drawn along beneath a solid load. The surface of the cylindrical bearing being 6 inches in width, transversely to the direction of motion, there is comparatively little tendency for the oil to escape, except by moving forward to the opposite side. The converse of this applies to the collar bearing 1 inch in width. The total length of each surface is 40·8 inches; the oil being supplied in four places, the length of each division is 10·2 inches. In the footstep bearing intermediate conditions prevail. The radius at the edge is

$1\frac{1}{2}$ inches. The mean radius describes 4·7 inches in a complete revolution, or 2·4 inches in a half revolution. Tests were made with four, three, and two radial grooves, the last forming one diametrical groove, which was found to give the lowest friction; possibly one radial groove would give a still better result, but it is better to avoid any departure from a symmetrically balanced condition.

Application of results of experiments.—The results of the recent experiments show that allowances based upon the older ones are ample under all practical conditions. They also show the good judgment displayed in the usual practice, wherein a heavy cylindrical bearing is carried in a brass step of large area and unbroken surface with the receiving edge chamfered off, to lead and press the oil against the surface, rather than to scrape it off.

Value of uninterrupted surface.—When a revolving shaft is supported in a brass which in one piece embraces one-half of the circumference of the shaft, and when such a bearing shows a tendency to heat, relief is obtained by reducing the brass so as to prevent contact at the two diametrically opposite parts of the brass. By this means the tendency of the brass to close upon the shaft and cause increased friction is avoided. But such a reduction of area should be effected with caution, or the oil may be unable to exist under fluid pressure of sufficient intensity, the metal surfaces will be allowed to come into contact, and the mischief will be increased rather than relieved.

Lubrication of footstep.—Heavily-loaded plane surfaces, either revolving as in an ordinary footstep, or reciprocating as in a cross-head slide, or a planing-machine table, are very difficult to lubricate efficiently. The footstep of an ordinary upright shaft 8 inches in

diameter can be loaded to 8 tons, or 400 pounds per square inch of bearing area, but must be carefully constructed and attended. The load should be carried by a solid gun-metal disc, of which the central third part of the surface should be cut away and the oil allowed free access to such space, from which it will run to the outside. If, however, the cavity in the foot disc is connected to a force-pump capable of delivering oil under a pressure of about a ton per square inch, the bearing will safely carry such a pressure. The overflow oil will be led to the suction of the pump, and the consumption of oil will be most trifling. An instance of this was given by Mr. Adamson before the Institute of Mechanical Engineers in 1885. In such a case the area of metallic surface provided for contact is a secondary matter, so long as it efficiently confines the oil against the bearing and under sufficient pressure. Though the area of metallic bearing may be very largely reduced with safety, so long as the supply of oil is maintained, yet, as this may at any time fail momentarily, and always on account of the necessity for support when at rest, such reduction should be made with caution, and pressures greater than 500 pounds per square inch adopted only from necessity. The oil used for this purpose should possess a high degree of viscosity, and also unctuosity, which must be imparted by a considerable proportion of organic oil in the mixture used. All heavily-loaded footsteps should be supplied with oil by means of a pump working at a suitable pressure. The cost of the instrument will be saved many times over in repairs, and the power expended in working the pump is utterly insignificant in comparison with that saved in friction of the footstep. The same remarks apply to collar bearings, whether upon horizontal or upright shafts. In these, however,

it may prove necessary to provide a stuffing-box, to prevent the escape of oil along the shaft, before oil can be supplied under pressure. In other cases also it is necessary to supply oil at several points through the solid block.

Plane reciprocating surfaces.—Plane sliding surfaces must of necessity possess a reciprocating motion. The initial friction experienced on reversing the motion of a shaft must also exist in these. This fact and the difficulty experienced in ensuring an efficient distribution of oil, combine to render inexpedient the application of a heavy load to such surfaces. In many such cases, pumps could be more largely adopted with advantage, but in others—as in vertical cross-head slides—such application is practically impossible. Plane sliding surfaces have probably never yet been loaded safely with more than 200 pounds per square inch, but with the adoption of copious lubrication this load may doubtless be much exceeded. It is, however, probable that it will be always impossible to apply to such surfaces a load equal to that which may be imposed upon cylindrical bearings, or at all events, unless special precautions are adopted to avoid any possibility of the instability of surface which causes increased friction on reversal of motion. There is no doubt that in well-scraped surfaces much of this unstable material has been removed, and the durability of such surfaces under hard work has been often found to be very great. Whether this durability depends upon the undoubted density of surface and freedom from fibre, or upon the attraction for oil possessed by the minute channels made by the scraping tool, may never be conclusively determined.

Application of oil-pumps.—Pumps for the supply of oil to bearings may be applied so that each one supplies its own bearing, or one pump may be used to supply a

series of bearings by the use of a high-level supply tank, with separate pipes freely supplying each bearing and a receiver below, provided with a strainer and with separate pipes from each bearing, the whole being plainly visible, to show at once if any stoppage occurs. Any bearing, such as a footstep, which requires to be fed at a high pressure, should be provided with its own pump, so that it may be supplied with special oil if necessary, and also so that the main pump will not be required to pump against a higher pressure than that due to the maximum difference in level between the pump, the tank, and the several bearings. For moderate heights of lift a rotary pump is best, on account of its compactness, facility of application, and uniformity of stream of oil. For greater heights of lift a single-acting force- or ram-pump may be used with an upward continuation of the discharge-pipe, to give an approximately uniform stream. For discharging under pressure a double-acting pump with two rams, or one of the differential class should be used. A single-acting ram-pump with an air-vessel may be used if the quantity of oil is large, but with a small quantity of oil this is likely to lead to oxidation. When an air-vessel is used, the air should not be changed more frequently than is necessary, as fresh air has a greater oxidizing effect upon the oil than air whose oxygen is partially exhausted.

Lubrication of bearing from below.—For the moderate supply of oil to a bearing of a horizontal shaft, a ring or chain is sometimes used to bring up a supply from a reservoir placed beneath the bearing, and into which the oil returns after use. In this case some means are required for spreading the oil, so that it reaches the surfaces requiring to be lubricated. Where a portion of the bearing surface at the bottom can be left open,

the oil-bath and pad as adopted by Mr. Tower—and as used in high-speed machinery—probably cannot be surpassed in efficiency. This is, however, obviously inapplicable to bearings heavily loaded on the under side.

Lubrication by siphon wick.—Well-fitted siphon lubricators are very reliable for drop lubrication, and have been used on all most important bearings until the recent introduction of copious lubrication. Each siphon lubricator consists of a vessel placed upon the cap of the bearing, or in some other convenient position, with a tube leading downwards to the lubricated surface, which tube is continued upwards, so as to prevent the oil from flowing directly down to the bearing. A loosely-fitted worsted wick is used to draw the oil over into the pipe by means of a siphon action. This wick is provided with a piece of copper wire for the purpose of withdrawing and re-inserting the wick each time the engine is stopped and started. If this is neglected, the oil passes over at an equal rate, whether the engine is at work or at rest, causing great waste. The wire also acts as a distance measure, to ensure that the end of the wick reaches to a lower level than the bottom of the oil-box, so as to give a good stream of oil. In the use of these lubricators no oil is returned to the oil-cup, and in too many cases the overflow from the bearing is wasted. In other cases it is collected for purification. In all cases the economical use of the oil, the cleanliness of the neighbouring parts, and the necessity for avoiding frequent re-filling of the oil-cup, call for judicious regulation of the supply to each bearing. This can be effected by an increase or decrease in the number of threads in the wick; but in no case should any regulation of the length of the wick be attempted. This should always reach the proper distance down the tube, or an occasional dry bearing

must be expected. These lubricators should be adopted for the piston-rods and valve-spindles of all engines, the cross-head slides of vertical engines, and in all positions where the drainage from a copiously lubricated bearing cannot be conveniently collected. Compound boxes are often provided in an elevated, easily accessible position, with a number of pipes leading to different surfaces, each fitted with a separate wick, and acting quite independently of the others. These should, however, be so divided that a special oil may be supplied to the piston-rods and valve-spindles.

Lubrication of crank-pins. — Crank-pins are often lubricated by means of a wick supply which delivers the oil at the required rate. In one arrangement the crank-pin is provided with a banjo, which consists of a cup so arranged that its mouth always occupies the same position opposite the centre line of the crank-shaft, and is carried by a tube mounted on the end of the crank-pin, and by which the drops received in the cup are transmitted to the crank-pin. A ring is sometimes applied — especially in double-throw cranks — behind the crank, in a manner precisely equivalent to that of the cup. In other cases the tube leading from the lubricating cup is so situated that a wiper on the connecting-rod end touches the tube end lightly, once in each revolution, and removes the drop of oil. This is a neat and excellent plan when well designed and fitted, and accurately adjusted. If, however, the wick is so adjusted as to give the oil slightly in excess, the drop may increase so rapidly as to fall away from the pipe just before the wiper comes round to pick it off, when the oil will probably be wasted, and the crank-pin allowed to run dry. A modification of this principle is used in gas engines, in which a wire is alternately dipped in oil and cleared by wiping across the lip of a

pipe, along which the successive drops are conducted to the point of application.

Lubrication of cross-head slides.—A comb or scoop of muntz metal is often used for the lubrication of the cross-head slides of a horizontal engine. Such a distribution should carry the oil in quantity throughout the full length of the stroke, or at least, much more than half-stroke. The proper application of this principle necessitates channels along each side of the slide of sufficient capacity to return all the oil to the basins at each end of the slides. In a horizontal engine running in the usual way, in which the pressure is upon the bottom slide, the whole may with advantage be surrounded with a lip sufficiently high to impound oil to a level just over the sliding surface. The capacity of the end basins should be ample, and the quantity of oil in use should be so adjusted that it does not wash over. Many horizontal engines work at a serious disadvantage, by reason of the fact that though the slides are efficiently lubricated at each end, where they are exposed to the least pressure, they are very feebly lubricated at the centre of the length, where they are subjected to the greatest pressure. Others again are so well fitted and lubricated, that they have shown the scraping marks quite freshly after working for ten years at a speed of 1000 feet per minute. A horizontal engine working against the upper slide is more difficult to deal with. A wick-regulated supply may be provided, in which case the edges of the cross-head block should be chamfered off to spread the drops, otherwise the drops will be likely to be cut off and diverted from the surfaces upon which they are intended to fulfil their purpose. Another plan is to cut a recess in the upper surface of the block, to be filled with oil, which is spread upon the surface by means of a cork float or

iron rollers kept up by springs; these are, however, less satisfactory.

Needle lubricators.—For ordinary shafting, lightly loaded, a system of copious lubrication with oil service and drainage would be too complicated, and would prove very costly in the first instance and in subsequent working. Needle lubricators are therefore employed, which are simple and efficient for the purpose. A glass vessel is used with the mouth placed downwards, pointing to the shaft, and stopped by a loosely fitting needle. When the shaft is at work a slight vibration is imparted to the needle; this, assisted in some cases by slight expansive force of the contained air in the vessel, due to increase of temperature, causes a sufficient amount of oil to be supplied to the bearing when at work. But no oil is passed when the shaft is at rest. The friction must, however, of necessity be very much greater than would be the case if copious lubrication could be adopted. Oil should always be applied at or near the centre of a bearing. It will then work towards each end and lubricate the whole length, which sometimes fails to ensue when two lubricators are applied at some distance on each side of the centre.

Lubrication by grease.—Grease or solidified oil is applied by a lubricator with a screwed cap, which may be tightened manually or automatically. If the temperature should accidentally rise, the grease becomes more fluid, flows down upon the bearing and arrests the heating. In ordinary work these lubricators require only the least possible attention. But when one has melted out, there is a risk that it may fail to receive any attention before the bearing suffers permanent injury.

Lubrication of steam cylinders.—Minute adjustment in the supply of cylinder lubricant is found to be of

very great importance. Many lubricators have been designed to effect a very small constant and positive feed by mechanical means. Others act by displacement caused by the accumulation of water produced by the condensation of steam. In most cases this water is supplied by a condenser, specially provided as an adjunct of the lubricator. In other cases the condensation is effected by the surface of the lubricator itself, as in Ramsbottom's lubricator of thirty years ago. Each of these may deliver its contents into the cylinder directly, under the adjustment of a cock, based upon trial. But lubricators of each type may be much more efficiently adjusted by actual observation of the oil on its way to the cylinder. For this purpose the oil is directed through a nozzle, usually pointed upwards and directed along the centre of a vertical tube of strong clear glass, and containing water, through which the separate drops of oil rise, as they form on the nozzle and become detached. The oil is led away from the upper part of the tube, to some point where it becomes diffused in the steam current. This is described as "sight feed" adjustment. In all cases in which water is used in the glass tube through which the oil ascends on its way from the nozzle, an objectionable stain of oil is liable to form upon the surface of the tube. This arises more frequently when crude heavy oils are used, but does not of necessity imply that the oil is otherwise objectionable. The stain may, however, be prevented by the use of a little soap in the water contained in the tube.

Self-lubricating surfaces.—Bearings have been used for special purposes which dispense with the necessity for oleaginous lubrication. Plugs of a different substance are inserted in the metal bearing, or blocks of such substance are fitted to take the place of metal bearings. Such substances are supplied by different

makers, and contain plumbago, steatite, and other material. They act by shedding a most minute film of slippery material over the surface of the bearing, which film appears to act as a lubricant, and to waste with exceeding slowness. In some cases neglected surfaces of iron or other metals shed a similar film of pure or oxidized metallic particles over the surface with a similar effect.

Lubrication by water of immersed bearings.—Bearings for the support of shafts or other revolving or sliding parts of brass—and occasionally of iron, where it will not be exposed to corrosive influence—are sometimes made of lignum vitæ. The largest field for the application of this principle is in the propeller-shafts of steam-ships, in connection with which the lignum vitæ is fitted in strips, the length of each strip and the direction of the grain being parallel to the length of the shaft, with spaces between for the access of water, which acts as a form of lubricant. But it is also used dry, when its action is very similar to that of the plumbago compositions. For special purposes other woods are sometimes used in bearings. They should, however, possess a greasy nature like lignum vitæ; or if this is absent, linseed oil may be forced in by pressure—or by pressure and heat—before fitting in position. For any kind of light work immersed in water, wood of almost any kind will answer best if fitted with the surface on the end of the grain. For iron shafts in water, white metal bearings, and also plain bushes of hard cast-iron are used. In almost every case in which a bearing works in water the pressure per square inch is very low.

Lubrication of wheel teeth.—The teeth of heavy wheels should be lubricated by means of a brush applied well down each tooth when the engine is

turned slowly. When lubricated by the application of a brush to the points of the teeth when working at full speed, the bulk of the grease is thrown off at once, and little if any reaches the point of useful effect. Old grease is dangerous to use, on account of the iron and other foreign bodies it is likely to contain. Wheel teeth should be regularly scraped down and examined.

Choice of lubricating materials.—In practice, the choice of lubricating materials available for any particular purpose is generally a large one, comprising oils and greases of animal, vegetable, and mineral origin. The natures of these vary exceedingly, as also the degrees of efficiency with which they fulfil their purpose.

Viscosity of lubricant.—The most obvious point of variation is in their palpable consistency or viscosity, which condition is reciprocal to fluidity. No absolute scale exists whereby the degree of viscosity may be clearly described. But an arbitrary scale of comparison may be constructed, based upon the length of time occupied by a definite quantity to pass through a certain aperture, the whole apparatus and contents being kept at a fixed uniform temperature. The quantity may be estimated either in weight or volume, but preferably the former. All oils and greases lose viscosity, or become more fluid as the temperature rises; but in some oils this change is much more rapidly developed than in others, so that while at one temperature a number of oils will stand in a certain order with respect to each other, at another temperature the order will be changed or even reversed. In measuring the viscosity of an oil for a certain purpose, the temperature should be first kept at the mean temperature to which it will be exposed in work, and afterwards at the maximum temperature. The relative viscosities of

different oils show their order of resistance to displacement in the bearings when in use at the temperature of the experiment. In all cases, a certain amount of this quality is necessary to prevent the whole of the oil from escaping under the pressure, and allowing the metallic — or other — surfaces to come into contact, causing their abrasion and ruin. This is especially the case with heavily-loaded bearings, but the tendency is present in a less degree in even the lightest bearings; and an appropriate degree of viscosity in the oil will prevent it. But the resistance to displacement which is thus useful, acts in opposition to all motion of the surfaces in contact.

Oxidation of lubricant. — The atmosphere causes changes in all oils exposed to contact with it. As a rule oxygen is absorbed, and the oil becomes thick and unsuitable for use for lubricating purposes. The activity with which this proceeds depends upon the class of oil, and upon the amount of surface in contact with air. A film of oil very subject to this action, if spread over any surface and left exposed all night, will be quite gummy next morning. When waste or similar material is soaked in such oil, so as to expose a large amount of surface to the action of the atmosphere, the absorption of oxygen becomes very rapid, and the temperature often rises so much as to cause the mass to burst into flame, which will set fire to any susceptible material in proximity to it. Mineral oils are very much less susceptible to this action than organic oils. A judicious admixture of mineral oil with an oxidizable oil, will practically prevent the oxidation of the latter, thereby increasing its working value, and the length of time during which it remains efficient, and also obviating the risk of fire arising from the use of such oils. An oil which is perfectly safe for use in one

room may become dangerous in another room of higher temperature.

Partial evaporation of lubricant.—At ordinary atmospheric temperatures very few oils suffer any appreciable loss of substance by evaporation, and serious loss arising in this way will usually be detected by smell. Cylinder oils should, however, be tested in this respect at the temperature corresponding to full boiler pressure, if saturated steam is used, or at the steam temperature if superheated.

Lubricating power of lubricants.—The absolute lubricating power of oils depends upon their unctuosity or greasiness. This property is useful in causing a reduction in friction, and it is possessed in very different degree by oils of equal viscosity. Appliances are made which measure the resistance to motion upon a bearing of constant dimensions, loaded with a given weight. The results obtained in their use are of value, taken in connection with viscosity, but it is doubtful whether the quality of unctuosity alone has been satisfactorily measured. It is, however, well established by experience and experiment that animal oils such as sperm, neatsfoot, and lard oils possess the property in the highest degree. Vegetable oils, as olive, castor, and rape oils, are somewhat inferior. Mineral oils are very inferior in this respect, and as a rule are unsuitable for use alone in bearings of any kind.

Application of lubricant.—Pure oils of any kind are generally very much less efficient for use than oils mixed for a particular purpose. The required conditions are that the viscosity shall be as low as will suffice to carry the load; the greasiness shall be as great as possible; and just sufficient mineral oil shall be added to preserve the condition of the oil. If this principle could be completely applied, the amount of

power to be furnished by the engine would be reduced to a minimum. Owing, however, to the great diversity in the conditions appertaining to different bearings, the number of separate oils required in one establishment would become very inconvenient, and it is necessary to classify the whole, so that while some bearings will receive oils not perfectly adapted to their conditions. yet, taking the system altogether, the slight losses which are incurred are more than compensated for by simplicity in application and freedom from liability to error. In some cases a very few bearings may be of importance sufficient to call for a special supply of oil, while others which may individually be of trifling consequence, may assume very great importance by reason of their large number. In the first case, the force of circumstances will probably attract sufficient attention to them, but in the second case there is a possibility that the value of an oil of low viscosity will be overlooked. Very great differences in the power required to drive a cotton mill have been observed to follow upon changes from one class of oil to another, used for the light spindles employed in spinning machinery. Here the load upon each spindle is so small, that it is scarcely possible to obtain an oil with true lubricating properties, whose viscosity is insufficient. A very small additional amount of viscosity will therefore cause an increase in the power required, without securing any benefit whatever. In *Engineering*, vol. xlvi., p. 288, a case is mentioned in which the power required to drive an American spinning mill varied 12 or 13 per cent. with the use of different spindle oils. The variation in the power required for the spindles alone must have reached a greater percentage. Mr. Longridge, in his report for 1880, also gives an instance in which the power required to drive

a weaving shed rose from 298 to 364 I.H.P., simultaneously with a change in the oil used, and fell to 300 on reverting to the oil formerly used. Such cases are far more common than is usually imagined, both in connection with spindles and shafting. Usually one class of oil will be required for all ordinary shafting, lubricated by wick or needle lubricators. Main shafting and engine bearings will generally require an oil of high viscosity, though in copious lubrication oils may be used of much lower viscosity than would be otherwise necessary. As the continuous stream of oil effects a reduction in friction, and a further reduction follows upon the use of thinner oils, there is every reason to hope that the practice will soon become almost universal in connection with heavy bearings.

Lubrication by grease.—Though in all cases the use of oils of the lowest possible viscosity results in an important economy in power, yet it often happens that the use of thicker oil allows of a reduction in the necessary amount of attention, the value of which exceeds that of the power thus lost. This point becomes developed to an extreme in connection with the use of solidified oils or greases. In many cases, the convenience, safety, and economy to be secured by the use of these are undoubted; but there is good reason to believe that, as a rule, the amount of loss in power is much underestimated. The advantage of cleanliness is, however, secured in an eminent degree. Owing to the chemical action of oils upon india-rubber, this advantage is of special importance in connection with the use of such belts as are largely composed of india-rubber.

Deterioration of oil in use.—Lubricating oils after a time in work lose their useful power. This may arise from what may be called a mechanical destruction of the particles, or from chemical change. In this respect

unmixed mineral oils are especially deficient. Suitable oil, which is well protected from the access of dust, will, however, retain its power for a very long time. Oil which is liable to oxidation should obviously be protected as far as possible from exposure to the air. A fine strainer should be constantly used to remove all gross impurities.

Decomposition of unsuitable cylinder oils.—Cylinder oils are exposed to high temperatures when in use, and hence all tests should be applied to them when at the same or a little higher temperature. At all steam temperatures, and especially at those corresponding to the high pressures now adopted, it is found that all oils and greases of animal or vegetable origin are decomposed with more or less rapidity into their constituent parts, of which the most objectionable are of an acid nature. These acid bodies immediately attack the iron or other metallic surfaces with which they come into contact, forming metallic soaps and destroying the structure, especially of iron bolts, and of such cast-iron as may be deficient in closeness of texture. Striking instances of this are often met with in connection with irregularity of form arising from lugs, feet, or ports cast upon the plain body of the cylinder. Some of the soap thus formed remains attached to the surfaces upon which it is formed, including the working surfaces of the cylinder, piston, rods, valves, etc., causing a very great increase in the frictional resistance; and the balance, along with the unchanged oil or free acid, is carried forward by the current of steam through the entire circuit of the several successive cylinders, the condenser, and air-pump, causing extensive wasting of all exposed surfaces with which it may successively come into contact. Serious mischief is thus caused in engines which are supplied with steam produced from fresh

water. But the mischief assumes much greater importance when a surface condenser is adopted, and the condensed water returned to the boiler. The boiler is thus exposed to the same destructive agency, and the accumulated acrid matter gives an increased action throughout the circuit. If a jet condenser is adopted and part of the air-pump discharge-water is fed to the boiler, an intermediate condition arises. The destructive action in the boiler is largely checked by the use of soda, but the only absolute solution of the difficulty is found in the discontinuance of the use of organic oils and tallow.

Use of mineral oil in steam cylinders.—Mineral oils have been largely used for some years in steam cylinders. The absolute lubricating power of these oils is distinctly inferior to that of tallow or organic oils, but is adequate to the work, and their use is necessitated by their chemical inactivity at temperatures corresponding to any practicable pressure of saturated steam, while many resist the decomposing action of superheated steam. It is not too much to say that without them the present high pressures of steam would have been practically impossible. As elsewhere explained, the presence of unchanged oil in the boiler is also objectionable, so that it is necessary to efficiency that the quantity of oil admitted to the engine shall not be appreciably in excess of actual requirements.

Mixed cylinder oils.—By the addition of organic oils to mineral oils, the lubricating power of the latter is increased. A small quantity added in this way is not found to be liable to injurious decomposition, and in some cases provides the means for a material reduction in the total quantity of oil supplied. The use of such mixed oil is of most importance in horizontal engines, and in those provided with slide-valves, on account of the greater friction prevailing in such cases.

Difficulty encountered in changing cylinder oil.—When an engine has been lubricated for a long time with tallow or other organic grease or oil, and the surfaces have become coated with the iron soap previously referred to, special attention is required for some time after making the change. The first application of the mineral oil brings the soap to the surface, and a most alarming and immediate increase in friction arises. If the steam is not condensed and returned to the boiler, a liberal increase in the amount of oil used, frequent cleaning of the pistons, and patient attention will soon surmount the trouble. If the steam is returned to the boiler, caution must be used as to admitting oil in excess; the boiler water must be frequently changed; and internal examination of the boiler made. The advantage to be secured by the change will, however, amply repay the trouble.

Specific gravity of oil.—Oils vary in specific gravity, and experts utilize this fact in separating or corroborating the identification of different oils. But the specific gravity of undoubtedly pure oils of the same class varies so much by reason of differences in raw material or processes of manufacture, and the total range of specific gravity of oils of all kinds is comparatively so small, that evidence thus obtained is of little use, and certainly of no use in the estimation of lubricating value. Thus pure samples of white whale, neatsfoot, lard, olive, ground-nut, and rape oils, may be found with specific gravity ·917. An oil whose specific gravity is ·880 may be either sperm or mineral oil. It also does not appear that even in comparing undoubtedly pure oils of any class the specific gravity gives any indication of value.

Efficient mixing of oil.—Mixing of oils should not be attempted except by those who possess proper

appliances for the attainment of a suitable and uniform temperature and also for the agitation of the mixture. If these conditions are not fully secured, the mixture is imperfectly performed, and the several components are liable to become separated again after a short time.

CHAPTER XLV.

CORROSION OF METALS.

Corrosive destruction of metallic structures.—Without any important exception, all the metals used in engineering structures are met with naturally in combination with other substances, of which the chief is oxygen. They all exhibit a decided tendency to return to such a condition, which tendency differs in degree in various cases, but which, if realized, usually involves the total or partial destruction of the structure, and must be resisted by suitable means.

Iron is especially liable to corrosion.—Iron is the metal which is most extensively used, and, of all the common metals, is most prone to waste by corrosion or oxidation. A clean surface of iron, exposed to dry air or to pure water, which contains no dissolved oxygen, or carbonic acid gas, will remain unchanged for an indefinite length of time. But if any vapour of water should be present in the air, it tends to become condensed upon the surface, by reason of changes in temperature. When this takes place, the condensed water absorbs carbonic acid gas from the air, and the two together, assisted by dissolved oxygen, act upon the metallic iron to form ferrous carbonate. This is decom-

posed, yielding ferric oxide, sesquioxide, or red oxide of iron, and liberating carbonic acid in a nascent or specially active condition, ready to attack another equivalent of iron. It is found that in the first stage of the cycle, the chemical action is much less active than in the second stage. In the second stage the action is sufficiently powerful to proceed in the apparent absence of free water. The influence of rust, already existing on a surface, can be shown on a penknife blade, by applying two similar drops of water, side by side, and allowing them to remain until rusting is fairly established. If then, one spot of rust is scraped entirely away, and the other partially so, it will be found that, in the former case, rusting is arrested, but not in the latter case.

Irregular corrosion arising or promoted by galvanic action.—When a surface of iron is exposed to water of any kind, except the very purest, and especially to sea-water, or any saline solution, a galvanic action is likely to commence at any point which presents the least departure from a condition of uniformity, such as a particle of copper or lead, or of mill-scale, or even a variation in the texture of the metal itself, or in steel in which manganese happens to be unequally diffused. Such galvanic corrosion is generally local in its action, and may be confined to a very small area, upon which it will raise a cone or tumour of rust, which will show distinct layers on cutting through, and which will leave a pit on removal. This action may originate in the presence of a very small speck of matter which sometimes remains unchanged, and is found in the centre of the tumour of rust. It may be noted that the thickness of the rust produced is about ten times as great as that of the metal destroyed. Iron and steel, or cast-iron and wrought-iron, or even two different

brands of material nominally the same, or two parts of the same piece, will act upon each other so as to promote corrosion, or to concentrate it into one part, a very good instance of which is often seen in a small wicket gate, or other article of forged iron, exposed to the friction of the hand, and also to the action of the weather. In such cases some parts or fibres of the metal are more susceptible to the weather than others are; and the thin film of rust, so produced locally, is imperceptibly swept off by the hand. This action, being continued over a number of years, shows the texture, and the working and welding of the iron, in a very curious manner. A similar action is produced in a greater degree upon anchors, chains, and forgings exposed to sea-water. In such cases, the harder and more crystalline metal is less subject to oxidize than the softer and fibrous metal; and when the two exist in contact, the action upon the more susceptible parts is increased by galvanic agency.

General corrosion.—Ordinary rusting or corrosion takes place with practical uniformity all over the exposed surface of iron when the substance is fairly uniform. But local and general corrosion are insensibly merged in each other. In either case, the close attachment of rust already formed is a powerful agent in extending the action. In the case of iron structures, tools, machines, etc., in constant use, under ordinary conditions, any rust which is produced is removed at once, either by actual abrasion, or by reason of the elastic movement of the metal; and probably this is the reason why such structures waste slowly, as compared with others, when both are exposed to equally damp air. But where a strong corrosive agency exists, such means are insufficient to check the action.

Corrosion promoted by mechanical action.—Where the

mechanical strains imposed, or those arising from changes of temperature, are so distributed as to take effect in a locally limited manner, the scale or rust may be only cracked, and not entirely removed, which action, long continued, causes the formation of a groove along the line upon which the metal is exposed by such crack. In boilers of all kinds this often leads to most serious loss of strength. Propeller-shafts also show startling instances of the same action. Thin riveted work, as girders and light railway trucks, also show it.

Corrosion promoted by a stream of water.—A jet of water at a high temperature, as in a fine leakage from a boiler seam, often causes rapid oxidation of iron, and the continuous immediate removal of the oxide produced, leaving a clean, well-rounded cavity and the metal quite bare.

Corrosion of iron and steel plates.—Iron and steel plate-work furnishes a very instructive field in which to study the phenomena of corrosion. Plates are delivered in the yard covered by the moderately hard "mill-scale" or magnetic oxide of iron, which, while it is perfect, furnishes an efficient protection to the metal beneath. But at the edges, where the plates have been shorn to size, clean metal has been left which soon acquires a thin coating of rust. Each plate is punched for riveting, and on all hands meets with violent treatment, so that the black scale becomes cracked, streaked, and starred all over, the whole of which lines and patches show out in red on the first wet day. When such a surface is painted over, and put into work, the scale becomes loosened, and drops off, carrying with it the paint. The paint remains only upon the parts where it was applied to the bare metal or to a trace of red rust. After the lapse of a certain time, varying in different cases, the mill-scale is certain to

drop off all surfaces exposed to damp. The metal below the scale becomes oxidized, commencing at the edges or at any existing crack in the scale; a little oxide is first formed, which presses off the scale slightly. The presence and chemical influence of the red oxide continues the process along the surface, the black scale also becoming partially changed to red by absorption of oxygen. This can be well seen in a vertical plate, which has been placed for some months in a quiet situation, and from which the scale can be peeled off in very large pieces.

Influence of mill-scale.—Mill-scale or black oxide of iron is always formed when wrought-iron or steel, at a red heat, is exposed to the action of the atmosphere. When devoid of water, it contains 27·6 per cent. of oxygen, as compared with 30 per cent. in the red oxide which is formed at lower temperatures. The thickness of mill-scale continues to increase until the metal becomes cold or is shut off from access of air. A thin film is a good protection for the metal beneath it, so far as chemical rusting is concerned, but is no practical use when resistance to abrasion is required. In the rolling-mill, plates are often marked by means of paint before they are cold, whereby the full thickness of scale is prevented from forming. When such plates are used in quiet places, the paint-marks and the thin scale beneath them are often found intact after the lapse of many years, and when all surrounding parts are in a highly corroded condition.

Partial dropping of mill-scale allows the development of galvanic action, due to the negative character of the remaining scale, and the positive character of the unprotected plates. By this means the absorption of oxygen of the parts which are electro-positive is very much promoted, and corrosion proceeds rapidly. The

actual amount of such action appears greater by contrast than it would if the whole were uniform. To obviate this evil, plated work should be exposed to the weather as long as possible before painting, so that the scale will naturally fall off, after which the surface may be cleaned and paint applied. When any scale remains after scraping, it should be removed by the application of a weak solution of sal-ammoniac, or dilute acid, after which the surfaces should be cleaned by the use of a weak alkaline solution, followed by a wash of pure water. All surfaces should be scraped clean before painting, and thoroughly dried, and especial attention should be paid to the neighbourhood of any blister, and to the removal of anything which appears to form a centre of excitement of galvanic action. When particles of this kind are in very intimate contact with iron, they are sometimes very difficult to remove. Their power to excite galvanic action depends not only upon their nature and their bulk, but upon the close, perfect character of the contact between the substances.

Influence of moist cinder in proximity to iron.—Any acidity in water which has access to iron causes corrosion to proceed with great rapidity, as also does ferrous sulphate (copperas), and other compounds containing sulphur, which are produced from wet heaps of slag, furnace ashes, and, in some cases, from coal. Pipes and other articles made of iron and buried in moist coal-ashes corrode with astonishing rapidity, and with such an absence of uniformity as to suggest that galvanic action has a considerable share in the process, facilitated by the conducting power imparted to water by ferrous sulphate. In one case, met with a few years ago, it was necessary to dig down to a chain land tie, which was found to be buried in ashes. Many adjacent links

were found to be absolutely corroded through at the welded end, while the opposite end of each remained nearly of the original size of three-quarters of an inch, though deeply scored by corrosion. Fig. 140 is taken from a photograph of an actual link. The wasted material was found to be of a bright green colour, but rapidly turned red by absorption of oxygen.

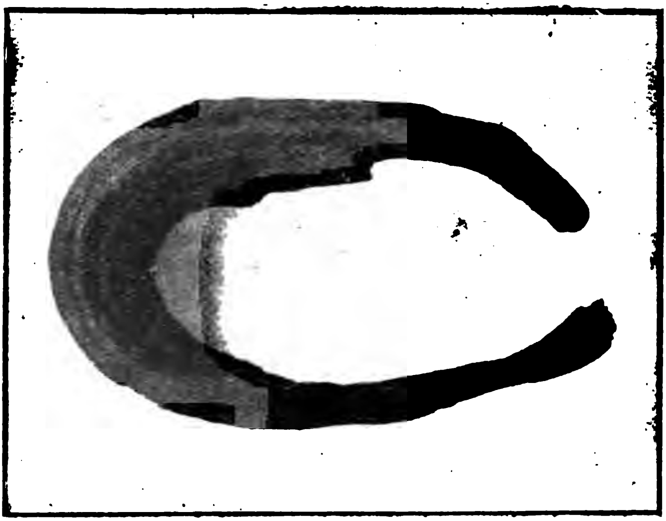


Fig. 140.—Link of chain, corroded by action of wet ashes.

Sulphurous corrosion.—Instances of wasting promoted by the joint action of steam and sulphurous fumes, acting upon metal at a moderately high temperature, have been often met with in the lower part of the front plates of boilers, where the pernicious practice of cooling ashes on the foot-plate by means of water has been allowed.

Influence of organic acids.—Grain, sugar, and other organic substances, by their decomposition (especially

at high temperatures), produce acid matters which possess most destructive power over exposed surfaces of iron. Unprotected iron, in contact with wood, often suffers from acidity derived from the wood. Oak timber is especially destructive in this respect.

Influence of alkaline substances.—Alkaline solutions have little or no injurious action upon iron, but they act powerfully upon copper or brass surfaces. Hence valves, &c., which are ordinarily made of brass, are made of iron in some cases, with the special object of securing greater durability.

Influence of high temperatures.—A high uniform temperature generally produces dryness or the conversion of all water into the form of true vapour, when such high temperature exercises a preservative action. This is also the case when such high temperature is the means of driving off oxygen or carbonic acid from solution in water, where otherwise these substances would act upon iron. But where such high temperatures do not prevail with uniformity, a distilling action arises by which water is condensed upon the cooler metal surfaces, giving rise to an exceedingly powerful corrosive action. At temperatures above 600° F., however, iron is oxidized with great rapidity. This action is shown in the uptakes and funnels of various kinds of boilers, and in superheaters. This takes place even when the surfaces are kept dry, but still more so when water is occasionally allowed access. High temperatures also indirectly promote corrosion in the cylinders of steam-engines in which tallow or oils of organic origin are used. A high temperature accelerates the decomposition which produces fatty acids, which combine with iron oxide to form an iron soap, involving extensive waste of the metal, especially if it should happen to possess, at any part, an open texture.

Influence of sea-water.—Sea-water is found to act more energetically upon iron than fresh-water does. In the cases of wrought-iron and steel, it does not appear that the action penetrates below the surface of the metal, though there it is energetic. But in the case of cast-iron exposed to sea or brackish water, the entire mass becomes soft and spongy, and loses its metallic characteristics, this first arising in any parts of the casting which were originally spongy. This action is due to the porous character of the metal and the conducting power of such water, in some cases the action being still further increased by acidity. It is more powerful when the water is warm or impure, and it proceeds with much greater rapidity when the hard skin of a casting is removed from the surface. In such waters a cast-iron air-pump corrodes most rapidly, though the condenser casting may be almost completely protected by the unbroken skin of the casting. The action may be also somewhat assisted by the pulsations to which the air-pump is constantly exposed in work.

Protection by skin of cast-iron.—The substance of cast-iron is little if any better than wrought-iron for resisting corrosion, and when porous may be much worse than wrought-iron. But when left unbroken, the skin of a casting, under all conditions, imparts great powers of resistance. The composition of such skin is essentially different from that of mill-scale, as it is formed by a combination between the molten iron and the sand of the mould.

Comparison of steel and iron.—Great difference of opinion is expressed as to the relative merits of wrought-iron and steel in resisting corrosion. In most cases in which well-made steel is used, in which the several ingredients are uniformly diffused, more substance will be found to be lost than with the use of wrought-iron,

for the reason that the action takes place with practical uniformity. But, as a rule, wrought-iron loses more strength, because the corrosion is more local in its character; especially is this the case in forgings exposed to salt or brackish water, in which the strength may suffer seriously, while the loss of weight is comparatively trifling. In either case, if scale should become enfolded in the mass during manufacture, the original strength is affected, but the original deficiency in strength is rapidly increased by reason of the abnormal corrosion thus caused. Steel is less subject to this defect than is iron. Indeed, in the former case there is scarcely an excuse for its occurrence. The precise relative liability to corrosion possessed by iron and steel is, however, a point of comparatively little importance. Both suffer severely if neglected; and in almost every case both can be protected by equally simple means, and a moderate amount of attention.

Influence of hardness of iron.—In all forms of cast-iron, wrought-iron, and steel, hard metal is much less prone to corrode than soft metal, the power of resistance appearing to be imparted by the combined carbon.

Corrosion of metals, other than iron.—Lead, zinc, tin, and copper oxidize in the same general manner as iron, but more slowly. In each case, when a coating of oxide is formed, it does not extend in the same manner as a coating of iron oxide, but exercises a protective action over the metal below. In some districts, however, acid fumes prevail, and act upon the oxides to produce soluble substances, which are removed by rain, causing a fresh exposure of metal, and the rapid destruction of the whole.

Protection of bright surfaces.—In machinery and some other work, permanently bright surfaces are desired. The brightly-polished condition may be pre-

served by protection against exposure to water or to moist air of a temperature above that of the metal surface, by which an active film of water would be precipitated upon such surfaces. Also when a film of oxide is formed upon the surface, in spite of all precautions, it should be immediately removed by friction. If this is delayed the operation rapidly becomes more difficult, and the removal fails to restore the surfaces to their original polished condition.

General protection of iron surfaces.—The great majority of iron surfaces may be efficiently protected by a coating of one or more of the following—(*a*) Metal of a nature less susceptible to corrosion than itself; (*b*) an adherent film of black oxide of iron; (*c*) paints and varnishes; (*d*) asphaltic coverings; (*e*) cement; (*f*) coatings of a temporary nature; (*g*) if no coating can be practically applied and reliably maintained, the surface may be brought into a galvanically negative condition.

(*a*) The metals used for the protection of iron are zinc, tin, copper, and brass of various kinds. Zinc and tin are applied by first removing all scale and trace of dirt, and then plunging the object into a melted bath of the coating metal. The process of covering by zinc is called "galvanizing," and is used most extensively for the protection of roofing-sheets and sheet-iron utensils, but it is also used for ironwork for the most diverse purposes. True galvanizing consists, however, of a deposition of zinc by means of a galvanic current, from solution. Probably this is the process originally adopted, whence the name. Either process is not properly applicable to details of ironwork which are not completely finished to size, shape, and surface all over, and in which holes are required to be subsequently pierced. Roofing-sheets are, however, usually

galvanized before corrugation, and thus become subject to slight cracks in the coating. But, by reason of their more perfect protection, they would doubtless well repay the additional cost incurred in applying the coating to each sheet after rather than before corrugation. The shell-plates of torpedo-boats and similar craft are galvanized with great advantage, as an amount of abrasion which would strip off any coat of paint or composition will not suffice to remove the coating of zinc. As an additional precaution such surfaces are, however, generally painted. In work of this kind, the rivets are of simple iron or steel. Any accidental scrape which strips the paint off a number of rivets, is sure to leave some of the zinc surface bare as well, in which case the rivets will be to a great extent galvanically protected, provided that the water does not contain acid or other chemically active substance in solution.

Tin plates are sheets of iron cut to certain sizes and coated or superficially alloyed with tin, in a manner corresponding to galvanizing. Tin is comparatively non-susceptible to corrosion, and the coating of tin plates is harder and better able to resist wear than either pure iron or tin. In the workshop the sheets are almost invariably cut to suit the work, in which operation the iron is left unprotected at the edges, but in the majority of cases these are re-protected by the solder used to make good the joint. When tin plates are otherwise applied in exposed situations, they very soon become dilapidated by corrosion originating at the edges, or holes inserted so as to leave the edges exposed.

Copper is usually applied electrolytically to surfaces of forms suitable for the purpose. Pump-rams and similar objects are, however, protected by forcing on a tube of copper which must possess sufficient strength to withstand the pressure necessarily applied in the oper-

ation. Brass is applied to rams or piston-rods by forcing, as in the application of copper. Turned objects of iron may be protected by a covering of brass cast in position. When, however, such objects are more than 3 feet in length, the longitudinal contraction of the coating in cooling is apt to draw the ends away from any shoulder which was in close contact while hot.

Galvanizing and tinning are obviously unsuitable for fitted work; and the coats will wear off under an amount of friction which varies with the thickness of the coating, but is never very great. When zinc is partially removed, the remaining zinc gives a weak galvanic protection to the bare iron for a little distance. But when tin is partially removed, the iron is destroyed more rapidly than if no tin were present.

(*b*) Iron is protected by a thin film of black oxide, under the Bower and Barff patents, by exposure of the cleaned surfaces to heat, and either superheated steam or an admixture of furnace gases. The coating allowed in any particular case is just sufficient to give protection, but not so thick as to become inelastic. The oxide protection imparted by these processes is much less adapted to resist wear than a coating of either zinc or tin, but for work exposed to very moderate wear the process is a good one.

(*c*) Paints consist of oils which dry and harden when exposed to the atmosphere, and to which various earthy substances or metallic oxides are added, to give body, or to impart some required colour. In most paints, either red lead or white lead or both are employed. In the process of boiling oils, litharge is used, which also imparts lead to the oil, so that lead is usually present in some form in paints. When the paint begins to shrink from the action of the weather, the oil resin shrinks away first, and the lead changes to oxychloride, and assumes

a crystalline form, in which each crystal acts as a centre of galvanic action, ready to act most injuriously upon the iron the moment the waste advances so far as to leave the metal bare. Therefore, whenever a lead paint is used it should be covered by a succeeding coat of a kind free from the same objection, or—still better—it should be scrupulously maintained in good condition.

Oxide of iron is largely used in paints, and is not liable to the same action as lead. But it often contains sulphuric acid, free and combined, in both of which conditions it possesses a destructive action upon iron. In this respect, all oxide paints should be tested by treatment with hydrochloric acid and chloride of barium, which will give a white precipitate in the presence of sulphuric acid. On the whole, this is a good paint, but the practical identity of the oxide with that produced in the rusting of iron is apt to lead to confusion of opinion.

Oxide of zinc is free from the objections applicable to lead and to iron oxide. It has been used largely, but not so largely as it deserves to be, owing to a mistaken—or, at all events, an exaggerated—impression that it is deficient in body.

Boiled oil sets or hardens by reason of a process of oxidation which results from exposure to the air. In the absence of a free exposure to the air this oxidation is prevented or retarded. In the process the oil is converted into a resinous substance which resists the destructive action of the atmosphere for a long time. The resin possesses acid properties which, in combination with the small proportions of lead in the oil, impart to the whole a slightly corrosive action. Many oils possess drying properties, but linseed oil is the one which is by far most largely used for this purpose. Boiled oil is often found to be seriously deficient in drying properties, so that, notwithstanding the fullest

exposure to the atmosphere, it remains in a soft and sticky or "tacky" condition. Paint in which such oil is used is never efficient as to resisting wear or simple exposure to the weather. Sometimes paint of this kind is covered over with paint of a better quality, in the hope to correct the fault. By this means the stickiness of surface is rectified, but the inner coat remains in a soft or semi-fluid condition. The outer coat in drying tends to shrink, and ultimately does so, breaking up into small pieces, and leaving gaps through which the inner soft coat is again visible. Usually the pieces into which the outer coat breaks up, and the spaces between, are very small and numerous. But sometimes the pieces are large, and the spaces between exceed an inch in width, showing that the inner coat possesses the defect in question to an extraordinary degree. Paints may be tested for adhesion and body by applying to a solid surface, allowing to dry, then dipping the article in cold and hot water, and testing with a hammer.

Oil varnishes are practically paints, in which resins or other substances soluble in oil are substituted for the earthy matters. They waste in the same way as paints, and are subject to the lead crystallization and acidity due to the oil. With this exception, good examples are satisfactory in use. They are chiefly used to impart a hard smooth surface after the application of paint.

Spirit varnishes are those in which the oil is replaced by a medium which is not changed or hardened by the action of the atmosphere, but which evaporates unchanged—or nearly so—leaving the resinous substances as a film on the surface. The evaporation of the solvent leaves the film in a porous condition from the first, and the pores increase with the wasting of the varnish. These pores may be more or less blocked by the application of a second coat of varnish, but, in a

longer or shorter time, they re-develop, giving access of moisture to the metal, and allowing the development of corrosion. In connection with metal-work, varnishes of this kind are chiefly used on account of their clearness and transparency, to protect bright surfaces of metal.

A few compositions are not acted upon by water, and they may consequently be applied to wet surfaces. But the large majority of paints and varnishes must be applied to dry surfaces, or they will be practically certain to peel off in large pieces. It is also very desirable that they should remain dry until set, so as to allow the freest access of air; consequently fine weather should be utilized for this work as far as possible. In all cases, compositions of the paint and varnish classes should be applied in thin coats. Two thin coats are worth much more than one coat of equal thickness, because they give a more dense covering, and the second coat fills the pores of the first. All traces of scale and dirt should be removed before the application of paint or cement. Dirt is generally supposed to be removed, but scale is often allowed to remain for insufficient reasons. Old compositions in perfectly sound condition, however, need not be removed. In all cases, special care should be taken to protect all surfaces which will be inaccessible after closing up. Every possible opportunity should be taken for the examination of surfaces which are only occasionally accessible.

(*d*) Asphaltic coverings consist of cements of bituminous substances, in combination with material to give increased bulk and strength. Limestone and calcareous matter generally appear to have an unexplained natural affinity for bituminous matter, and are generally found to take an important part in compositions of this class. These compositions are often required to fill a considerable thickness, and coke and

other porous substances are used to reduce the weight. This is only wise when it is possible to ensure that an ample and definite thickness of solid asphalte shall exist around every piece of porous material. This is a difficulty, and any defect in this respect renders the mass pervious to water which is allowed to reach the iron. Corrosion ensues, and the composition is lifted, without showing any clearly visible signs on the surface. These compositions, however, give good protection when applied hot, to hot clean surfaces, and in accordance with the above conditions.

Artificial asphalte is very inferior to the natural material in respect to density and toughness of substance, and power to resist wear. Gas tar enters very largely into the composition of these substances. Tar is very good for application to surfaces after the manner of paint, and allowed to dry. But it is very deficient in power to harden, when in mass, as in asphaltic mixtures. For application to surfaces, tar is improved by the addition to the fresh tar, undeprived of its naphtha by distillation, of a little tallow to reduce its brittleness, and fresh quicklime, to overcome acidity and give body to the whole. Though in all cases the application of heat is an advantage in the use of tar compounds, such is the affinity with limestone, that marks made upon it with cold tar remain good after the lapse of twenty or thirty years. Dr. Angus Smith's composition is used chiefly for pipes. It consists of coal tar, pitch, 5 to 6 per cent. of linseed oil, and sometimes a little resin. This is heated to 300° F., and the pipes after thorough cleaning are heated to 700°, placed in the composition, and allowed to remain until cooled to 300°. The tar and pitch should be quite free from naphtha and naphthaline.

(*e*) The cement used for the protection of iron is

chiefly of the Portland class. A wash of neat cement, mixed with water, is good under almost all conditions where the work is not exposed to much abrasion, and the colour is not objected to. Such a coating of pure cement is practically impervious to water, but when in considerable thickness is apt to crack in setting. Sand or other material is therefore added to reduce shrinkage, by which step also the cost is reduced. The latter fact, however, often leads to excessive addition of such aggregates. Silicious matter mixed with Portland cement binds the whole together, and to the surface of iron, and prevents porosity in the mass when not used to excess. A coating of cement and clean sand in equal parts will resist a large amount of wear. Crushed brick is also used to mix with cement for this purpose, but soft brick and all traces of mortar should be rejected. Old fire-bricks, free from foreign matter, are very good. Porosity is sure to exist if the cement should be insufficient to fill all the interspaces throughout the mass. If any porosity should exist in the presence of various saline and metallic substances in solution, these will obtain access to the surfaces of the iron and produce chemical action, which will lift the cement from the iron, showing little or no sign on the surface. Cement is also broken away by blows, strain, or great vibration. But with these limitations cement is an excellent protection to iron plates, when properly applied in thicknesses of three-quarters of an inch and upwards. A thin wash of cement applied to the cleaned surfaces, before the application of the composition, very much assists the adhesion.

(*f*) Coatings of a temporary character are used to protect bright work when in transit or store, or when in process of erection in a position temporarily exposed to damp. For this purpose a mixture of tallow and white

lead has been mostly used. This is very efficient for a few days, and is easily removed when required. But after the lapse of a little time the white lead dries the tallow and forms a film, which allows the access of water at some point or other, followed by the shedding of the film, leaving a rusty surface beneath. At the present time a thin spirit varnish is mostly used. This holds better than tallow and white lead, is more elastic, and is easily removed when required by means of turpentine or other solvent.

(*g*) The destructive action which is developed when metals of different natures are placed in galvanic contact with each other, and with a fluid capable of exciting a galvanic current, is turned to good account by placing zinc in contact with iron. Zinc is used because it is electro-positive to iron, and therefore while it wastes the iron is saved. This system has been largely adopted with success in marine boilers, but in fresh water is less successful. It is, however, now very largely displaced in favour of chemical protection by the use of soda, which neutralizes the destructive acids found both in sea and fresh waters.

Self sealing of iron surfaces by corrosion.—The expansive force of corroding iron is useful in protecting the threads of bolts and other opposing surfaces, which are exposed to the action of water. Corrosion takes place for some time after a bolt is placed in position, but the oxide which is formed constitutes a tight packing, which excludes air and water, and preserves the thread surfaces. It is thus often found, that while an old bolt is quite efficient as it stands, there is not sufficient thread upon it to hold the nut in any different position.

Tightness of rust-joint.—An ordinary rust-joint depends for its efficiency upon the expansion of corroding iron. Iron borings, which are really powdered cast-iron, are

mixed with a substance—usually sal-ammoniac and sulphur—which causes rapid oxidation. But before the oxidation is completed, the mixture is rammed into the required position. The gradual expansion which takes place then causes the application of pressure in an exceedingly perfect manner, to ensure the tightness of the joint. Any agency which causes the removal of oxygen will cause the ruin of a rust-joint.

Corrosion in sulphur-joints.—Sulphur cement, used for fixing iron-work, is apt to cause sulphurous corrosion of the iron, which also is accompanied by expansion, which often causes damage, as—for example—in stones in which sockets are cut to receive iron posts, &c., secured by this composition. These are often found to be split across by the expansion of the iron in its combination with sulphur.

CHAPTER XLVI.

FORCING AND LIFTING APPLIANCES.

Use of forcing-screws.—Nearly every detail about an engine, and many machinery details, should be arranged to be taken apart by the help of forcing- or lifting-screws. To receive these, holes should be tapped in the cylinder covers, valve-chest covers, and junk-rings, which holes should be closed by set-screws when not required to be used.

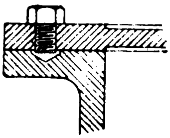


Fig. 141.—Tapped hole for forcing-screw, closed by set-screw when out of use.

A forcing-screw is used for separating two surfaces, by turning the screw when its point bears against the back part. The screws should tightly fit the holes, so as to prevent the accumulation of dirt in the threads, as dirt is apt to jam the screw so that it may be wrenched off even without bearing against the back part. Set-screws for stopping tapped holes should project through the front plate by about one thread, the back plate being drilled quite clear, as in Fig. 141.

In the removal of any cover which may by any possibility be exposed to pressure at the back, all nuts should be slacked a turn, before any are slacked further. The cover may then be started, with safety, whereas if some of the nuts are removed, the cover may be violently blown off. By the same means, the cover may be kept quite square and fair in removing by means of two screws only, though for large covers three or four screws are better.

Use of lifting-bolts and straps.—Tapped holes are also provided to receive eye-bolts for lifting details. As far as possible these should be so placed that the eye-bolt will stand quite vertically and accurately over the centre of gravity of the object. Sometimes plain straps are secured to the detail by set-screws for lifting, and in other cases permanent handles are attached.

Selection of sizes of bolts.—The provision of a large selection of eye-bolts or forcing-screws is quite unnecessary. Usually, $\frac{1}{2}$, $\frac{7}{8}$, and $1\frac{1}{4}$ inches are most suitable sizes, and for very heavy work $1\frac{3}{4}$ may be used. Long eye-bolts are sometimes necessary, but they are more liable to bend in use than short ones, which should therefore be used as far as possible.

Use of forcing wedges.—In the absence of forcing-screws, wedges and chisels may be used for removing covers. This practice should, however, be discouraged, and never allowed except where proper chases are cut in which to start the wedges.

Rope tackle.—A small hand or luff-tackle is almost indispensable for light and moderate lifts; it should consist of two blocks, each containing three pulleys or sheaves, through which a cotton rope is rove. The blocks should be critically examined in every part, as to uniformity and sufficiency of strength. The hooks should be capacious, made of highest quality Yorkshire

iron, and made to swivel. The neck of the swivel is the part which above all others in a block is apt to break, and this should receive special attention. It should be well rounded in the shoulder; its minimum sectional area should be such as to avoid the imposition of a greater stress than $3\frac{1}{2}$ tons per square inch; and its centre line should point directly down towards the suspended load, or up to the point of suspension. The bearing area of each sheave upon the pin should suffice to allow 1 square inch per half-ton in load upon the two parts of rope leading to the sheave, and the tackle will work much more freely if the load does not exceed one-third of a ton per square inch upon the length $\times$ diameter of pin in sheave. Solid gun-metal sheaves running upon steel pins are best, but large sheaves may be bushed with gun-metal. For light loads or side-pulls, the fall of the blocks may be hauled upon by hand, and this is the most efficient manner of applying hand power where strength is available. As a rule, however, a crab, loaded to prevent sliding or overturning, must be used for heavy lifts. If the prevention of sliding depends entirely upon weight, the amount of this will be not less than from two to three times the tension upon the horizontal rope led to the crab. The liability to overturning of the crab is greatest when a horizontal rope is led over the barrel, and least when it is led under the barrel.

Chain-blocks.—For vertical lifts, one of the many patterns of differential or worm-wheel chain-blocks is often convenient. When they are in good order they are safe, but they are extravagant in power, and it is often difficult to bring a sufficient number of men to bear upon the work, while the application of a crab is usually impossible.

Overhead travelling crane.—As a convenient means

for lifting details, always available in an engine-house the moment it is required, some form of overhead travelling crane is beyond comparison the best. Its simplest form is seen over a vertical engine, where a single stout bar, or a pair of parallel bars or rolled joists runs over the cylinders. A pulley-carriage can be run to any part of the length, carrying a simple lifting-screw, by which means a cylinder cover, or a piston, can be promptly and safely dealt with. For deep lifts a hand tackle may be attached to the same gear, and for other lifts permanent eye-bolts are conveniently fitted, or holes are provided to receive movable eye-bolts.

In an ordinary engine-room—especially one which contains a horizontal engine—a conveniently arranged overhead travelling crane is often provided to command any part of the work. Preparation is made in the walls of the engine-house either by means of a cornice or corbels, to receive rolled joists along each of the greater sides. These joists are placed at such a height that their upper surfaces are about 3 feet below the ceiling or roof. Rails are placed upon the rolled joists to support a carriage, upon which the transverse rails are laid which support the crab, which is thus adapted for travelling in either direction. Many kinds of crabs or hoisting-winches are made, but probably the one which best combines simplicity and efficiency, comprises essentially a chain-barrel, worm-wheel and worm, mounted upon their respective shafts, and actuated from below by means of ropes. The lifting power of this crane should be sufficient to deal with the largest cylinder of the engine. If any part of the bed-plate exceeds this weight, it may be wholly or partially lifted by extemporized means. The lifting of cylinders by any such means is, however, difficult, inconvenient, and, in many cases, dangerous. The propor-

tions of the crab should be such as to allow the rapid lifting of the largest cylinder cover by two men. Heavier loads may then be dealt with by more men, and the use of a snatch-block. Such a crane can be used for any stonework which may be deemed necessary in the foundation. Stonemasons, as a rule, however, do not approve crabs worked from below by a rope; and hence, at the time when foundations were built throughout of ashlar stone, preparations were sometimes made for the use of a handle crab upon the ultimate longitudinal and transverse joists, the requisite headway being obtained by postponing the erection of the roof or upper floor.

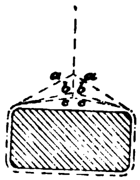
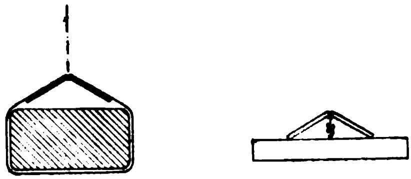


Fig. 142.—Load lifted by encircling chain or rope.

Slinging of loads.—In lifting various objects, a buckling-chain or a rope is usually employed to encircle the load, and the crane-hook attached to the girdle. Rope will yield elastically, so that it is able to adjust itself more or less under an excessive strain, but chain is comparatively inelastic, and many accidents have been caused by imposing an excessive load upon it, whereby the chain has been broken or the sides of the load crushed in. In Fig. 142 a load of 5 tons is assumed to be lifted, so that the tension upon the single chain or rope, above the junction of the two parts, will be 5 tons in each case. If each part of the buckling-chain occupies one of the positions shown in Fig. 142 by *a a*, the stress upon each part will be 5 tons. If the positions occupied are as *b b*, the stress upon each part

will be 10 tons; and if as in *c c*, the stress will be 20 tons. In each of the three cases, a $\frac{7}{8}$ inch crane-chain will suffice, and in the first case will also suffice for buckling-chain *a a*. In case *b b* the latter would require to be $1\frac{1}{4}$ inches; and in case *c c*, $1\frac{3}{4}$ inches. This is clearly shown by the parallelogram of forces and by experience. When lifting an angular load by means of a buckling-chain, packings should be interposed between the chain and the sharp corners, otherwise a chain should be used of a size greater than would be necessary on account of direct strength. The actual tension upon the chain may be obtained from the angle of the chain as measured by means of an ordinary two-



Figs. 143, 144.—Manner of estimation of tension on encircling chain or rope.

foot rule beneath the chain, as in Fig. 143. The rule is then placed against any straight-edge, as in Fig. 144, and the distance *x* measured to the centre of the joint-pin. The tension upon the chain $= \frac{6 \times \text{weight of load}}{\text{distance } x \text{ in inches}}$. The tension is also the same in the part of the chain beneath the load, provided that it is able to slip freely round the angles.

Annealing of chains.—Chains for important use should be occasionally annealed by fire. They should be slowly raised to a low red heat, and covered up by some substance capable of retarding the dispersal of heat, so that several hours shall be occupied in cooling.

Every link should be afterwards carefully examined all over, especially at the welds.

Working strength of chains.—A convenient and reliable rule for finding the safe load upon a good short linked chain, not exposed to shock, or passed round sharp corners, is to take the number of eighths of an inch in the size of iron, square it, and cut off one figure. Thus the safe load upon a $\frac{7}{8}$ inch chain is found to be 4·9 tons, which practically agrees with the usual load of 5 tons. Chains for the Admiralty are subjected to a test load of 12 tons for 1 inch chain, and other sizes in proportion to size. Chain of very good quality will break with $2\frac{1}{2}$ times the Admiralty proof load. If greater strength is obtained, the chain will probably possess low ductility, which is more serious than want of great breaking strength. Chains are made of same sizes of iron and same strength, but of less weight, by making the links longer and providing with studs to prevent bending of links. Table XXXIII. gives the several loads upon chains, and the approximate outside dimensions of links in short-linked chain.

Strength of hemp and cotton ropes.—The breaking strength of a hemp rope of 4 inches circumference may be easily remembered to be about 4 tons, and that of a cotton rope rather more. A rope of either kind, when in good condition, may be loaded to one-fifth part of its ultimate strength. A flexible steel-wire rope, of quality to pass Lloyds' test, and of 4 inches circumference, will carry about 32 tons before breaking. In this case also a factor of safety of one-fifth may be allowed. In a pair of three sheave-blocks, as usually arranged, the load is carried on six parts, and, allowing for resistance, the total load upon the six parts is about five times as great as that upon the part which is most heavily loaded. Thus a six-part tackle

TABLE XXXIII.—SHORT-LINKED CHAIN OF HIGH QUALITY.

Size.	Safe load.	Admiralty test.	Ultimate strength.	Approximate outside dimensions of links.	
				Length.	Width.
inches.	tons.	tons.	tons.	inches.	inches.
½	1·6	3·0	7·5	2·45	1·75
⅝	2·5	4·7	11·7	3·06	2·19
¾	3·6	6·8	16·9	3·68	2·63
⅞	4·9	9·2	23·0	4·29	3·06
1	6·4	12·0	30·0	4·90	3·50
1⅛	8·1	15·2	38·0	5·51	3·94
1¼	10·0	18·8	46·9	6·13	4·38
1½	14·4	27·0	67·5	7·35	5·25
1¾	19·6	36·8	91·8	8·58	6·13
2	25·6	48·0	120·0	9·80	7·00

will safely carry the amount of load which would just suffice to break one part. Before taking a hemp or cotton rope, except a new one, out of store, for any purpose in which it will be required to support a heavy load, it should be well examined, and a few fibres drawn from the end to ascertain whether they possess their original strength. When such a rope is almost fully loaded, it should be examined from end to end. If the several strands be equal it is probably safe, but if one strand projects further than the others, such strand is certainly weak, or the rope is in a "stranded" and unsafe condition. Ropes should be stored so that they will not appear in a sodden or dusty condition, as in either of these cases the strength will deteriorate.

Wire ropes.—Wire ropes are much used for special lifts, and are particularly well adapted for very high lifts, as they can be obtained in long lengths. For

many purposes they are much more handy than soft ropes. They should always be kept on a reel when out of use, as they cannot be coiled down on a floor in the same way as soft ropes, without giving trouble by "kinking" or "snarling." This is because a turn is put into or taken out of a rope for every coil laid down, according as it is laid in one way or the other. Similarly, when a rope is received from the maker it must not be unwound from the coil, but should be placed on a turn-table and spun round, so that the rope comes off precisely as it would from a drum. The turn-table need not be elaborately constructed, any strong door-

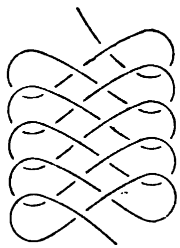


Fig. 145.—Rope arranged on floor in a manner to prevent kinking.

frame, or cover of square or circular form, will answer perfectly, with a nail driven in the centre, and spun upon an iron plate. The use or neglect of such an appliance, or some equivalent one, makes all the difference between a rope working without any vestige of trouble on the one hand, or on the other hand becoming entangled in kinks at every possible opportunity, while in the latter case the rope suffers materially in strength. When a reel is not available, and a wire rope must be laid out on a floor, it should be ranged in figures of 8, as in Fig. 145. The tendency of ropes to become kinked may be studied with advantage with the help of a few yards of flexible india-rubber tubing. Though

the trouble arising from kinking is much greater in connection with wire ropes than with soft ropes, the latter would benefit largely by reason of attention paid to the question. Wire ropes, when out of use for long intervals, should if possible be kept in lime-water. Wire ropes expand and contract by temperature, but not by decrease and increase of wetness, as soft ropes do.

TABLE XXXIV.—LOADS UPON ROPES AND DIAMETERS OF PULLEYS.

Circumference.	Cotton ropes.			Flexible steel-wire ropes.		
	Safe load.	Breaking load.	Smallest pulley.	Safe load.	Breaking load.	Smallest pulley.
inches.	tons.	tons.	inches diameter.	tons.	tons.	inches diameter.
1½				·8	4	7·5
2	·22	1·08	4	1·4	7	10
2½	·34	1·70	5	2·4	12	12·5
3	·49	2·44	6	3·6	18	15
3½	·66	3·32	7	4·9	24	17·5
4	·87	4·34	8	6·4	32	20
4½	1·10	5·49	9			
5	1·36	6·78	10			
6	1·95	9·76	12			

Loads upon ropes.—The loads applicable to different ropes are given in Table XXXIV., which also gives the minimum diameters of the respective pulleys which may be used for lifting tackle. The diameter of pulley for a cotton rope of given size is very much less than that for a wire rope of the same size; but the diameter of pulley for a given load is practically equal in the two cases. A barrel round which consecutive turns of rope are wound should be not less than one-fifth greater in diameter than the pulley which may be used

for the same rope. In all cases the diameters of pulleys and barrels should be made as large as conveniently possible.

Multiple ropes.—In many cases an amount of strength is required beyond the power of one part of any rope available, and several parts are required. These should be laid parallel and close to each other, and should be carefully watched, so that the tension is at every moment uniformly distributed. Many causes lead to variation in tension, such as a slack knot, or variable slipping. Any one in charge of important lifts should keep the most constant attention upon ropes and chains, and should be able to detect any irregularity at once by sounding. In many cases the slacking of a rope implies the overloading of another one; in other cases it indicates a probability that the load will overturn. In either case it should be considered as a sign of great danger.

Cranes assisting each other.—A load is often required to be lifted, the weight of which is beyond the power of any one crane or pair of sheer legs available, and two must be used. Two hydraulic cranes, each fitted with efficient relief-valves, in free communication with the cylinder, may be thus used in almost any way, to raise or to lower the load in perfect safety. Two steam or hand cranes may be freely used for lifting, but if both are directly attached to the same point upon the load, and if one is lowered more rapidly than the other—in the least degree—the slow crane takes the whole of the load, and an accident may be caused. To avoid such danger the points of attachment should be selected so that neither crane can be overloaded without an improbable amount of tilting taking place; and in some cases the condition should also cover the support of the whole by one crane, with the opposite end resting upon the ground. The

use of a snatch block, with one crane lifting at each part, will also ensure the loading of each crane equally both in lifting and lowering. The same principles apply to cases in which a lifting-jack is used to assist a crane.

Lifting-jacks.—The lifting-jacks in common use are the bottle-jack, Haley's jack, and the hydraulic-jack. The bottle-jack is exceedingly firm and safe for short vertical lifts, but is not convenient for pushing in a horizontal or oblique direction. A Haley or hydraulic-jack is just as firm as the last when lifting on the head of the screw, but is a little apt to slip when lifting on the claw. The use of the claw is, however, essential when lifting an object from a position near the floor. The bottle-jack is the least liable to derangement by reason of careless treatment, and the hydraulic-jack is most so. The hydraulic-jack is exceedingly convenient for use in confined situations, owing to the limited motion of the handle. All should be kept clean when out of use, and hydraulic-jacks should be occasionally pumped up and left raised for a time, to keep the leathers in good supple condition.

Wooden packings are usually adopted for the temporary support of boilers, cylinders, and other objects. Many accidents have been caused by the collapse of a set of packings which appeared to be secure. These often arise from the rounded faces of the packings, which become developed by shrinkage and wear of the wood, by reason of long-continued use. All timber packings—and especially old ones—should therefore be crossed in alternate layers when piled up to a moderate height. As a rule, three square wood packings of any kind should never be placed parallel and one over the other. The top packing used for the support of a rounded object should be formed as a cradle, to prevent rolling. This is best cut to shape, but

chocks may be nailed on a suitable beam. In lifting vertically a heavy object, especially by means of jacks, the packings should be followed closely to the work, so that the object cannot fall appreciably in case of accident. In a long lift by crane this is impossible, and all persons should stand clear from below, unless there is the utmost assurance as to the safety of the gear.

CHAPTER XLVII.

ENGINE AND MACHINE FOUNDATIONS.

Conditions to be fulfilled.—Large engines and machinery must be supported in such a manner that the truth of all working lines and surfaces shall be maintained. In many cases it is also necessary that vibrations caused by the motion of the several parts shall be quenched, and not transmitted to neighbouring buildings or structures. Much strength and rigidity are usually contributed by the several parts of the engine or machine itself. But this is seldom sufficient without assistance from the ground. In the absence of a rock foundation, it is usually difficult to secure absolute freedom from settlement. But in ordinary cases a slight amount of settlement or subsidence is of little importance, so long as it proceeds with uniformity over the whole of the surface in connection. When this is the case no distortion of line or surface will take place. Hence the importance of loading soft ground with a uniform weight per square foot, and of carrying the whole to a uniform depth, whenever the load upon a foundation is large. When the load becomes excessive, it may be treated as in the case of a chimney foundation.

Ashlar stone foundations.—Ashlar stone was used in engine foundations, almost to the exclusion of all other materials, until about twenty years ago. Each stone was squared all over and level bedded, lime mortar being used for bedding and jointing. Where oil is carelessly allowed to soak into the mass, it often happens that the stones work loose. This action is rendered probable when the bedding of the stones is defective, and especially when they are convex, so as to bear in the middle, or when they bear so heavily at the edges as to splinter. Irregular distribution of mortar is almost precisely equivalent to original irregularity in the surface of the stone. Any weak points in the foundation will render it more susceptible to damage by reason of irregularity or vibration to which the machinery is subject in work.

Brick foundations.—Brickwork has been largely used to replace ashlar stone, chiefly on account of its low cost, and the facility with which it may be erected to any required form, and without the use of cranes or moulds. The cost of labour and of supervision is much less with brickwork than with ashlar. Cement is used for the bedding and jointing of brickwork, and the bedding is solid to a degree which is almost impossible with large stone surfaces. Brindle bricks or blue vitrified bricks may be used when great strength is required, but care is necessary to ensure that the cement shall adhere in all cases.

Concrete foundations. — Concrete made of broken stone-ballast and Portland cement is now used almost exclusively. The labour involved is much less costly than with either ashlar or brickwork, and with proper precautions the work is at least as sound as any other kind. The stone-ballast or aggregate must be clean, and of mixed size when density of concrete is desired.

The cement must stand a test of 350 to 400 pounds per square inch in seven days after mixing, the test-blocks or briquettes having been exposed to the atmosphere for twenty-four hours after making, and in water for six days. A lower strength than this shows that the cement is bad, while a higher strength indicates liability to deterioration. The sand must be free from dirt, and above all things must not be used too liberally, as over-sanding has probably ruined more concrete than any other cause. The sand and the cement together form a mortar which binds the separate pieces of the aggregate into one mass. The interstices of the sand measure about 40 per cent. of the whole, so that 40 volumes of fine powder would just suffice to fill 100 volumes of sand, and the mixture would only occupy 100 volumes if the mixing could be perfectly performed. But such perfection of mixing is practically unattainable, and 50 volumes of cement are found to be necessary for 100 volumes of sand, or, as usually expressed, the mixing proportions are 2 to 1. In gravel and broken stone of approximately equal size, whether large or small, the interstices measure from 33 to 40 per cent. of the whole. More stone is contained in a cubic foot when the spaces between the stones of largest size are filled with stones of the next smaller size, and these again with stones of still smaller size, and so on until sand is reached. Concrete composed in accordance with this principle and carefully treated is perfectly safe under a load of 7 tons per square foot. Solidity imparts strength and also water-tightness, so that one condition may be taken as evidence of the other.

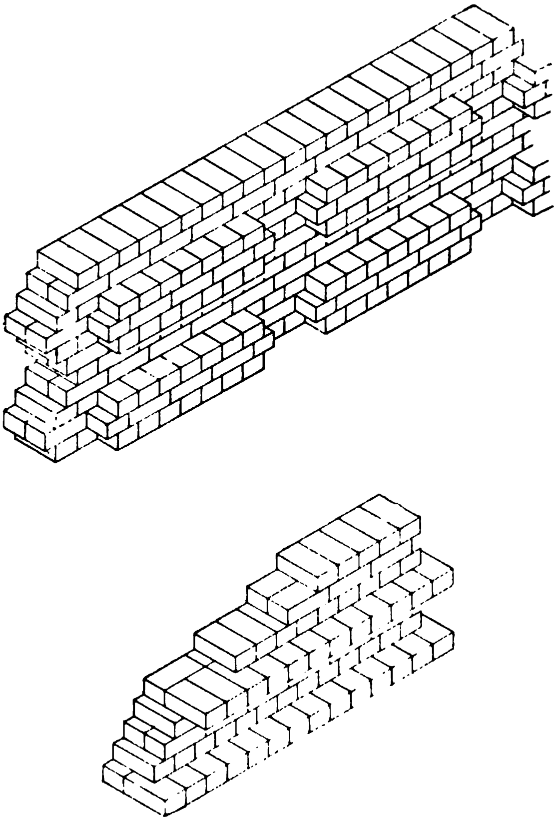
Moulds for concrete.—In most instances, concrete is moulded to the required form by means of boards or planks, planed to a good surface, and rubbed lightly

with soft soap to prevent adhesion of the cement. When new timber is used, it will be found on stripping off that each knot has left a light brown mark on the concrete, but this is of no consequence.

Facing of brickwork.—The use of wood moulds may be avoided by building the face of a concrete foundation, and also around each opening in brickwork, in cement, about 2 feet in height at one time, and allowing to set. These should all be bonded into the concrete in heights of two or three courses. A space of thickness greater than one course of brickwork should always be left between the several bonding blocks, or good sound tails of concrete cannot be secured. Alternate blocks and spaces of three courses, as in Fig. 146, are best, but they may be arranged as in Fig. 147, The blocks may be continuous, or interrupted by small spaces. Ashlar stones are sometimes built in the wall, for the same reason. In either way an excellent result is obtained at moderate cost.

Reduction of weight by means of voids.—Occasionally, in very soft ground, it is desirable to limit the load upon the earth, while great strength in the foundation may be superfluous. In such cases broken stone, sand, and cement are used in such proportions that very considerable spaces are left between the stones; the amount of cement and sand allowed being such as to just fix the several stones together, and impart strength to the mass in approximate proportion to the area of attachment. In this way a saving in weight may be secured of from 15 to 20 per cent. With a similar object, spaces may be left in ordinary concrete by building in wooden boxes or empty cement casks; and in any kind of foundation the same result may be attained by means of open cavities. The saving in weight thus secured will often be found to make the

difference between a firm and a yielding foundation in clay ground, while the cost will be rather diminished by the adoption of either means.



Figs. 146, 147.—Back of brick wall for bonding to concrete foundation.

Block-stone distributed through mass.—In depositing concrete in the mass, a great saving is secured by the use of "displacers," or clean rubble stones of "handy"

size. As nearly as possible without measuring, these are placed from 3 to 5 inches apart, so as to be well separated, and each completely surrounded by concrete. While the cost is reduced by this means, the quality of the work is fully maintained. It is generally considered that the use of rubble in this way reduces the liability of concrete in very long pieces to contract in cold weather to such an extent as to crack across.

Mixture of concrete, and contact with water.—All concrete must be well mixed. The amount of water to be added depends upon circumstances. Under ordinary conditions this should just suffice to bring the whole to a paste, which will naturally fill all corners of the mould when cast in by the shovel. But where solidity and strength are of special importance, the materials should be mixed so that they just appear to be dry when placed in position, but will show water all over the surface when hard rammed before setting. Concrete well mixed in this way is remarkably dense, as proved when required to be cut away after setting. It however requires to be kept well watered until it is completely set. Concrete of all kinds suffers somewhat if allowed to become parched during the time of setting. No amount of still water in contact with it will cause any injury. But if it is exposed to water at one side under a greater pressure than that at the other side, a percolation through the mass becomes established, and sound concrete is impossible. Means should therefore be taken to prevent any such pressure from arising. When a bed of concrete is laid upon wet ground, the first thin layer should be of dry stone without sand or cement, so as to give drainage. As the work proceeds, and weight is applied upon the concrete, the drainage spaces become filled up, allowing the whole work to settle slightly. Any springs which can

be discovered should be piped; and generally, means should be taken to allow the escape of every drop of water under pressure. A small amount of water under a very low pressure is sufficient to lift and destroy a thin horizontal sheet, or a vertical wall, of concrete; and still less is sufficient to cause damage by removal of cement, either from the whole mass or along the lines of least resistance.

Lime concrete.—Concrete mixed with hydraulic lime instead of cement has been largely used, but at the present time there is little or no saving secured, the work is not so good, and the practice is falling into disuse.

Securing loosened ironwork.—Occasionally, a wall-box or bracket is found to work loose. This may arise from some defect in the original work, but is quite as likely to be caused by excessive vibration arising from defective toothed wheels, or other similar cause, which should be at once remedied. In some cases the brick- or stonework requires to be renewed. In other cases, the work may be effectually secured by grouting with neat cement, mixed only with water, and fed through channels made for the purpose. If time allows, all crevices should be stopped at the face by cement pointing, a day or two before the grouting is performed. Otherwise clay, yarn stopping, or wood fillets must be used, and the face pointing applied after the removal of the temporary stopping. In any case the stopping must possess strength sufficient to resist the hydraulic pressure of the grout due to the head. The grout should be run from a point near to the highest level of the defective part, and the operation performed with the greatest possible degree of expedition. The grout may be fed in by a rod of wood or iron, which should be continued so long as any more can be induced to

enter. The last portion may be run in by the help of a pipe, to provide the necessary head. If desired, this may be done by a separate operation, when the first portion of grouting is partially set. Vent must be allowed for the escape of air at the highest point, for which purpose it may be necessary to drill a hole in the wall or in a casting, or lead a small lead pipe to tap an upward cavity. Sand should never be used in grouting, except in very small proportions, and only by those who are familiar with the use of cement. In any case, great difficulty will be experienced in completely filling any openings of a less width than half-an-inch. Cement grouting applied as above should be allowed to set for four days in warm weather, or six days in cold weather. Heat will accelerate the operation, but should be applied with great caution. The setting of good cement may be hastened by mixing with warm water; but though this has been found quite successful, it cannot be generally recommended, on account of the facilities for abuse which are presented. Work executed in ordinary concrete of proportions 5 to 1 will require at least three times as long for setting as is required by grouting. Stronger concrete will require less time than weaker, and for this reason it is often wise to allow a larger proportion of cement in cold weather than would be allowed in warm weather. Bolts may be most efficiently fixed in stone by means of cement, care being taken to ensure the complete filling of each socket, and allowing ample time for setting before applying any strain to the bolt. The load imposed should not exceed 300 pounds per square inch of cemented surface of bolt.

Plaster-of-Paris.—If sufficient time cannot be allowed for the setting of Portland cement, plaster-of-Paris may be substituted. This is weaker than Portland cement, and is slightly soluble in water, so that following upon

its use it may be necessary to take the first opportunity to rake out all accessible parts and replace by Portland cement.

Sulphur cement.—Sulphur cement, made from melted crude sulphur and sand, applied hot, is a most convenient material for filling spaces. When applied at the most suitable temperature, which is nearly as high as it will bear without firing, it is much more fluid than cement grout, and more completely fills the spaces. It has been much used for fixing bolts and stanchions in stone sockets. For permanent use in contact with iron it should, however, be avoided, on account of the strong chemical affinity between the sulphur and the iron, which leads to the destruction of the latter, and to the splitting of the socket, unless the surrounding material be of very great strength. In a flat surface, beneath a bed-plate, it is not so objectionable, as the effect of corrosion is merely to cause a lifting of the whole, generally to an almost imperceptible extent. Cast-iron resists the corrosive action of sulphur better than does wrought-iron, but in all damp situations, the use of sulphur should be avoided.

Limitation of vibration by elastic foundations.—In connection with machinery of moderate weight, working at a high speed, a foundation may be required for the purpose of quenching vibration. This it may do by sheer force of weight, or by its elasticity, which allows the absorption of the vibration before it reaches the ground. Elasticity of this kind in the foundation also improves the running of the machinery. A slight amount of elasticity is obtained by the use of pine, beech, elm, or raw hide soaked in oil, interposed beneath the bed-plate or feet of a machine. Asphalte may also be used with great advantage, in thickness not less than half-an-inch. In many cases, this may be best arranged

by the use of a flat cast-iron plate, fitted on the asphalte, then heated and slowly drawn down by means of the permanent bolts. In such a case adequate measures should be taken to prevent the spreading of the asphalte, when exposed to great pressure and to heat from any source, even from the weather. Broken stone or gravel may be used to reduce the tendency to flow in hot weather, but this involves corresponding sacrifice in the elasticity which is sought to be secured. Stonework may be filled up with asphalte, but there is considerable uncertainty with respect to the complete closing of all joints.

Elasticity is also obtained by the use of a wood foundation, carried by a layer of hair felt in a wood box. A case was lately referred to in *Engineering*, in which a thickness of 14 inches of felt was successful in this way. When this course is adopted, bolts are required to pass through the felt. The number of these should be as small as possible; large washer-plates should be used at each end; and a good thickness of hair felt should be interposed beneath each washer-plate.

Wooden foundations.—Foundations of wood are often used in temporary work. These are very conveniently fitted and secured, and are most efficient in use. They are, however, apt to become saturated with grease, even to a greater extent than they would in permanent work. They also become very dry, and their use involves great risk of fire.

Holding-bolts.—Bolts are always required to hold an engine to the foundation. These may be lewised or cemented into one or more courses of masonry, or into a corresponding depth of concrete or brickwork. But in this process great care is required to prevent inaccuracy, by which the correspondence of the work and

the bolts would quite fail. Accidents also arise by which bolts are broken, when the work of replacement involves the removal of the bed-plate, and is then a matter of great difficulty. The bed-plate must be placed in position by lifting over the bolts, and after the bolts are inserted it cannot be rubbed to mark the stone for fitting. On the whole, therefore, the practice of providing long bolts, accessible at the bottom by means of hand-holes, is greatly superior to that of fixing immovable bolts.

Base of foundation and preparation for holding-bolts.—The base of the foundation should be spread over a sufficiently large area for the weight applied. This was formerly secured by the use of a sheeting course of stone flags or landings, upon which the positions of the holding-bolts were marked. Hand-holes were marked off to leave about $7\frac{1}{2}$ inches on each side, built about 12 inches in height, and covered by stone blocks drilled for the bolts. This system is still good practice, but the sheeting course is usually replaced by concrete, for the sake of a reduction in cost. The hand-holes may usually be raised to shorten the bolts if desired, but the elasticity of the bolts as against shock is reduced in proportion to the length. They should not be less than 6 feet in length below the top of the foundation, and in ordinary cases it is not absolutely necessary to provide more. The foundation of concrete, brickwork, or masonry may therefore be carried up in a solid mass to about 7 feet below the top surface, when the positions of the holding-bolts may be laid down, the hand-holes built in brickwork, or mould-boxes fixed in position, and the building of the foundation continued for about 12 inches further. Cast-iron plates are then placed to cover the hand-holes, bolt-holes being provided in the plates. The top faces of the plates should be provided with bosses, to strengthen

the holes and to secure the wood boxes, which are used to preserve the openings for the holding-bolts. Rough wood boxes for this purpose are easily secured in correct position at their upper ends by means of battens nailed to them, but they should be tested for accuracy from time to time. Old iron pipes are sometimes used for this purpose instead of wooden boxes. In ordinary cases, the plates over the hand-holes should be provided with round holes of ample size to meet minor inaccuracies. But if after completion the hand-holes will be inaccessible, the bolts will be required to be placed in position as the work proceeds; the holes in the plates should be square, to loosely fit around bolts with square necks. An additional advantage is secured if the bolt-heads are of a spherical form in the bearing part, so as to allow a little freedom of movement, without throwing the strain on one side. When the hand-holes are accessible, the bolts should be provided with cotters at the lower ends, as they are more easily secured or loosened. The work should also be designed to allow the bolts to be passed down through the bed-plate when the latter is in its proper position. When the bolts are required to resist a great pull, the hand-hole plates should be strongly ribbed, for the sake of strength. The holding-bolts should be of uniform section throughout, to resist tensile stress, so that uniform elasticity is secured in the body, the screwed portions, and in way of cotter-holes.

Top surface of foundation.—The foundation in concrete, brick, or stone is built up round the bolt-hole boxes until the required level is reached. In some cases a course of ashlar stone is placed on the top. This gives a better surface than concrete, but the advantage is not a vital one. When a top course of stone is provided, the bolt-holes are marked off, after

the stones are laid, and from the bed-plate itself, if possible, then drilled through with accuracy. In the finest work the top course of stone is laid slightly above the required level, and dressed to suit the bed-plate, which is rubbed over it to indicate the high places. An alternative plan which is much more rapidly put into execution, and is less costly, is to finish the surface about half-an-inch below the bed-plate, which is adjusted in position by the help of wedges and screws, then packed up all over by ramming of cement and sand in about the proportions of 1½ of the latter to 1 of the former, mixed with water to give good consistency. If this is neatly and carefully performed, and under favourable conditions, the result is excellent. The wedges and screws should be left in position until the packing has fully set. The load upon such surfaces should not exceed 100 pounds per square inch, including all weights and tension of holding-bolts. Deflection of bed-plates should be regarded, and liability to vibration of any kind prevented as far as possible.

When the top of the foundation is of concrete, and forms part of the floor of the room, a special mixing should be used for the last two or three inches. This facing should be protected from trampling upon until fully set. For the first two or three days it must not sustain weight or pressure of any kind, but afterwards it may be protected by boards in contact, upon which persons may walk or stand. The general surface should be adjusted for drainage of any drip or slop which it may receive. It will be kept more easily and perfectly clean if slightly lower against the bed-plate, than if any slop is free to run over the whole of the surface. The most durable and smoothest surface is obtained by a mixture of Portland cement and shingle from a sea-beach. The cement wears away and leaves the smooth,

hard pebbles of quartz and other materials projecting. These resist wear and accidental displacement with singular success, and give a durable but very slippery surface. A suitable admixture of broken slag from a blast-furnace gives a surface which is much less slippery and fairly durable, though distinctly less so than one in which pure shingle is used. Crushed granite is very good for use in concrete floors. It is more costly than either of the mixtures first described, but in its general properties it occupies an intermediate position.

Cement stains on surfaces.—Workmen who use cement are very prone to leave unsightly stains upon surfaces of stone, brick, or iron which are extremely difficult of removal after being allowed to set. The majority of such stains may be easily avoided in the first instance, but in every case they should be completely removed before setting.

Support of floors.—Foundations should be prepared for supporting the adjoining floors, which should be liberally provided with lifting-plates, where any details such as air-pumps, bearings, etc., may at any time require attention. Iron plates are referred to in the chapter on boiler-houses.

CHAPTER XLVIII.

DESIGN OF ENGINES AND MACHINERY.

Plain design and use of curves.—The design of high-class steam-engines tends to become plainer with each change. The several details are designed throughout with the intention of securing most efficient fulfilment of their respective duties. Extremely little addition is now made, with the intention of improving the appearance. But in good practice the curves which are essential to strength and efficiency are carefully harmonized

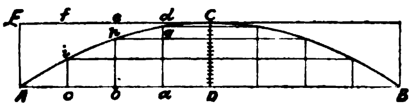


Fig. 148.—Production of approximate circular arc upon given chord and versin.

with each other, and with adjoining surfaces or lines. The more pronounced curves are generally approximate ellipses and parabolas, constructed according to one of the many methods described in geometrical works.

Use of circular arcs.—Many of the flatter curves, when really constructed by parabolic methods, are practically coincident with circular arcs. On this assumption, one of the latter may be plotted, when the centre is too far distant for convenient use. Divide the half-chord A D (Fig. 148) into any number of equal parts, of

which four are adopted in the figure. The points thus obtained are marked a, b, c in figure. The versine D C is then divided into equal parts, of which the number is equal to the square of the number into which the half-chord is divided. E C is drawn parallel to A B, and $a\,d$, $b\,e$, and $c\,f$ parallel to D C. $d\,g = 1$ part, $e\,h = 4$ parts, and $f\,i = 9$ parts, and so in progression by squares according to the number of divisions adopted, the 1, 4, 9, etc., divisions being taken from those upon the versine C D. In Fig. 149 the two symmetrical

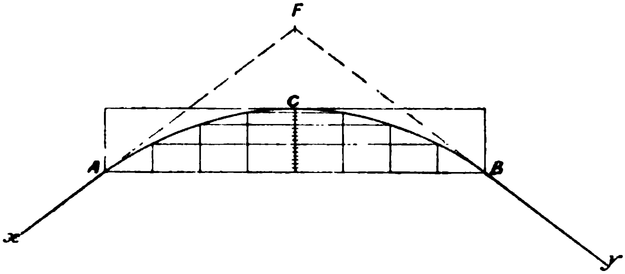


Fig. 149.—Production of approximate circular arc joining given lines.

lines X A and B Y may be supposed to require joining by a circular arc, to which each line is tangent. The two lines are continued to their point of intersection in F. If one circular arc is possible, the distances A F and B F will be equal. Points A and B are then joined by a straight line or chord. The point C is obtained equidistant from the two given lines and the chord. The curve is then completed as in Fig. 148.

Proportions of circular arcs.—The radius of a circular arc $= \frac{\text{chord}^2}{8 \times \text{versin}} + \frac{\text{versin}}{2}$. In flat curves the last element is seldom significant, and may be neglected when the versin $= \frac{\text{chord}^2}{8 \times \text{radius.}}$ Similarly, the chord $=$ $\sqrt{8 \text{ radius} \times \text{versin.}}$

Obsolete ornamentation.—Moulds, fluted columns, carved bases, and other additions formerly considered essential, weaken the work by irregularity of substance. Such details do not belong to the domain of the mechanical engineer, but have been imported from the architecture of a sunny climate, in which marble and other rich materials were employed to give a splendid effect. But the extensive imitation of these in cast-iron can only be considered as a failure.

Stiffening of bolt-holes.—When a bolt is required to pass through a flat plate, as a rule the latter should be increased in thickness by a facing, which stiffens the hole, even though the general strength of the plate is in no way deficient. Such a facing is also useful in chipping or planing the surface to fit the nut or washer. As a rule the height of such facings should be about equal to one-half of the diameter of the bolt, and the diameter somewhat greater than the washer used beneath the nut. Cutting out is sometimes adopted in connection with oblique bolts, in which case substance should be added behind the plate to make up strength, and avoid cracks through thin places.

Uniformity of substance.—Uniformity of thickness and treatment in all material is essential to uniformity of elasticity, and consequent advantageous disposal for strength. But such uniformity is of still greater importance in connection with material which is brought to shape by the assistance of heat. Irregularity leads to irregularity in cooling, and to internal strains, by which the strength of the whole is much reduced. Cast-iron is especially liable to weakness in this way, if the necessary conditions are not fully observed. Thick parts retain heat longer than thin parts, and continue to contract after the latter have ceased. If by reason of their form the latter are unable to follow, a crack is

produced, probably at once, but possibly after the lapse of many months, and without any apparent cause arising at the time.

Rounding internal angles.—In castings which are required to possess great strength, all internal angles should be filleted, even if it should be necessary to machine them out after casting. When uniform thickness in the casting can be secured, as in Fig. 150, it is impossible to make the radius of the internal fillet too great for the benefit of the casting. But when the outside cannot follow the inside, so as to maintain uniformity of thickness, the fillet may be hollowed to a radius from two-thirds to the full thickness of the plate, as in Fig. 151. Beyond this, the substance in the

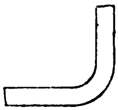


Fig. 150.—Curving of casting at angle.

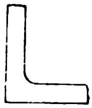


Fig. 151.—Curving of casting at angle.

angle is so far increased that irregularity of cooling becomes injurious.

Rounding of external angles.—The rounding of external or salient angles is only little less important than that of internal ones. If, however, the radius outside the casting is made in accordance with Fig. 150, the flat surface on the outside would often be unduly encroached upon. The difficulty encountered in making sound joints in wooden patterns at a reasonable cost would also be much increased. An outer radius equal to one-fourth or one-third of the thickness should, however, be allowed when possible. If the outer angle is left quite square, the metal will crystallize in two distinct parts, leaving a line of incipient division along the mitre plane, as in Fig. 152, and probably a slight

depression at some little distance from the corner. Such a defect is increased by an increase in the radius of the inside fillet.

Smoothing of arrises.—The external angles of fitted work should also be slightly rounded, not according to any rule, but just sufficient to mitigate accidents, and to reduce the indentations which they receive by reason of slight blows.

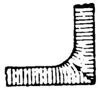


Fig. 152.—Curving of casting at angle.

Internal angles in fitted work.—Internal angles, such as in shafting at places where the diameter changes, or where shoulders or collars are applied, should always be well filleted. Cotter-holes or slots are almost invariably rounded at each end. This is done by machine, whereby the cost is reduced, and the perfection of fit and working strength of the work is much increased. Stress imposed upon a sharp angle is concentrated upon a very small portion of the material, while in a rounded angle, under otherwise identical conditions, the stress is more diffused, and damage to the material is very much reduced. Obviously, when once a little surface damage has been inflicted, it will extend with comparative rapidity. A sharp angle, being especially prone to extend into a fracture, may be itself almost regarded as an incipient crack. A specimen broken close to a sharp angle, not rapidly, but as a result of many years' work, exhibits a fracture with a great similarity to a clean cut, which is never imitated in any other case except by heavy transverse loading upon sharp-edged detail.

CHAPTER XLIX.

CHOICE OF PLANT.

Object of investment.—In the selection of plant for a new installation, or for the entire or partial replacement of an existing one, a leading object with the purchaser is to secure an ample and uniform return upon the capital invested, and for the vigilant attention bestowed upon the work.

Efficiency of service.—The value of the services rendered by the plant varies with the efficiency of the work, both as regards its obvious or intrinsic value, and the promptitude and reliability with which the work is executed. In almost every case, this efficiency is the first point for consideration. Many other and obvious commercial elements enter into this side of the question, but need not be here dealt with.

Total amount upon which interest is to be charged.—Interest should be calculated upon the total sum invested, considered at its lowest market value as a secured investment. This includes the cost of machinery, buildings, and foundations for the support, shelter, protection, and often for the advantageous display of the whole, also all fittings and appurtenances of the whole, and the total amount under all the succeeding heads which may

be expended before the whole becomes productive. In many cases the cost value of the works should be debited with the current value of works displaced, on account of their still possessing some working value though partially obsolete. In many cases—perhaps the majority—the items of cost and depreciation of buildings, and sometimes of plant, are consolidated and covered by a terminable rent, paid by a tenant to an owner of such works.

Depreciation.—The second item is depreciation, which is the difference between the value of the plant at the beginning and the end of the period in question. This should be so adjusted that at any time during the life of the works they may stand in the capital account at their just value. This is not necessarily that which would be produced in a forced sale, whether by auction or private contract; but such a cost as would be agreed upon between a willing buyer and an equally willing seller; or by a judicially-minded valuer acting between the two. Depreciation covers all damage by weather or the effect of time, that caused by wear and tear, and also any reduction in value which arises from extraneous causes, such as a change in the surroundings of the works, affecting markets, water rights, purity of water, light and air, supply of labour, cost and convenience of transit, etc., and the legal vindication of rights. Conditions may also arise by reason of the institution of improved means of communication, and in other ways, by which the item of depreciation may be partially transformed into one of improvement. Many of these items apply to the site, rather than to the works themselves, but those which apply to the works should be so graduated that their value will become extinguished by the time at which such works are due for replacement by new. If this is not done, the value of the old works

should be written off as a loss, but often such value is charged as a grievous burden upon new works.

Labour.—The third item is that of labour of a directly productive class, and that which may be necessary in the supervision of the same, or the clerical service, or the protection and watching of the same.

Stores.—The fourth item includes the cost of the whole of the consumable stores, including fuel, water, lighting material, oils, and sundry stores, and also the transit of materials for treatment in connection with the special business of the works.

Maintenance.—The fifth item includes the direct cost of repairs, whether effected by the workmen of the establishment or by outside contracts, including the whole cost of labour and stores consumed. Also, in case the ordinary duty of the works is interrupted, a proportion of the continuous charges upon the establishment should be debited to the account of such repairs. Such interruptions of work usually entail direct and indirect costs or losses in connection with the discharge of contracts. Such losses are also really part of the cost of the operations in question.

Insurance and contingencies.—The sixth item is that of insurance, which covers all losses by reason of exceptional occurrences of any kind, such as fire, explosion, accidents of any kind to plant or workmen, theft, suspension of work by reason of labour disputes, trade losses, and many other matters. This includes not only the premiums paid to insurance companies, but costs incurred for the sake of safety, direct and indirect costs and losses which arise from time to time and cannot be met by outside insurance, or which it is preferred to deal with otherwise, but which cannot be escaped.

Final balance.—The total cost and liability under the whole of the above heads and subdivisions must be

regarded in connection with the prospects of a profitable return upon the capital invested, or proposed to be invested, in any particular instance. If the margin is insufficient to cover all these, a loss will ultimately develop instead of a profit. A complete analysis of the facts as affecting each item is necessary before a well-considered decision can be arrived at upon an important question of investment. Obviously, it is most desirable to keep the first item as low as possible, without a preponderating sacrifice in other directions. One of the first questions to arise is the probable amount and severity of the work which will be imposed upon the structure in question, as this will in a great measure decide whether the depreciation of plant will arise chiefly through natural decay, or through wear and tear.

Examples.—In a rolling-mill or brickworks, the plant is subjected to very great wear and tear, and to little natural decay; consequently it must possess an exceptional degree of strength, but it requires comparatively little protection from the weather. In a timber-yard, it may be necessary to provide a large saw-frame for exceptional use. This would require to be protected by a building quite wind and weather proof. If a separate engine should be required to drive such a machine, cheapness of construction would be of more importance than an exceptionally low consumption of steam, or of fuel. A large lathe is in any case a costly tool, and where it is required for full work, a strong substantial tool to take a heavy cut will secure the best return upon the outlay. Also the indirect pecuniary interests of the owner of such a lathe may render necessary the greatest despatch in its use. But in another direction it often happens that the necessity for a large lathe is an imperative one, even though there may be every certainty that the tool will stand idle for the greater

part of its time; and in such cases there is often a perfectly legitimate call for a lathe of low original cost, even at the sacrifice of some considerable degree of efficiency. In many cases a gas-engine is the most economical motor which is applicable, notwithstanding the high cost of fuel in the form of purified gas, the chief advantage consisting in the low cost of labour. Insurance is sometimes directly of importance, but chiefly indirectly, as when appliances for the reduction of fire risk are provided, with a view to secure a reduction in premiums paid for insurance. Many other measures are adopted for the sake of preventing accidents which would interfere with work. This is the most powerful reason for the adoption of rope gearing, in spite of the loss of power involved in its use. Stoppages due to accident usually occur absolutely, or nearly, without warning, and the consequent interference with work and contracts is usually of greater consequence than the direct loss, even though that may be great.

Changes in practice and estimation of merit.—It is quite unnecessary to argue upon the advantages to be secured by a strict adherence to the rules of good, well-established practice. But cases constantly arise which cannot possibly be met in this way, and others which can thus be only partially met. New ideas are also constantly arising, and are laid before those who are interested, claiming advantages of various kinds, of which perhaps the most common is the saving of coal to the extent of 10 or 20 per cent., or even much more, and giving instances where such degree of success has been actually secured. The first question to be raised is in connection with the merits of the principle itself, as if it is of the perpetual motion order, or if antagonistic to generally-accepted physical laws, it may be at once rejected. However, the possible validity of the claim being accepted, the next

question is as to the conditions of the case in which success has been secured. Are the original conditions precisely the same in the two cases? Also, in making the change to the new principle, was any change made beyond those absolutely necessary? For instance, it often happens that a change is made in connection with a steam-engine which involves a stoppage of some considerable duration. Advantage is taken of such a stoppage to put in order everything connected with the engine, boiler, shafting, and in many cases even the machinery driven by the shafting. The flues are cleaned and repaired, and air leakages stopped, and often a sweeping improvement takes place throughout the work. Naturally all these measures have a most salutary effect upon the economical result, but the whole is often inadvertently credited to one particular point, which apart from all others may or may not be a desirable one. This is an exceedingly common case, but perhaps it will be more clearly put by assuming two mills to be built and fitted throughout in precisely the same manner, so that the original coal consumption is identical. One mill may be maintained in thoroughly good order from the beginning to the end of its life. But the other may be neglected, so as to fall behind, in respect to coal consumption, to the extent of 20 per cent. Its defective condition being noticed—though misinterpreted—it is decided to adopt some means, such for instance as the replacement of toothed gearing by ropes, and during the stoppage necessary for this purpose everything is put into the same order as the first mill already enjoys. Rope gearing is known to absorb more power than toothed gearing to the extent of 5 to 10 per cent. upon the total power of the engine. The new arrangement should therefore show an improvement to the extent of 20 per cent. due to the general repairs, cleaning, adjustments,

etc., and a loss of say 5 per cent. due to the adoption of ropes. This gives a nett advantage of 15 per cent., which at first glance might be claimed as entirely due to the adoption of rope gearing, but a complete comparison with the neighbouring mill will show that an actual loss has been incurred by the change of plan. From this instance is seen the vital necessity for the separation of all element of change, so that the exact effect of each may be estimated; otherwise data in connection with changes only suffice to mislead. At first glance it would appear that a rigid scrutiny would act as a deterrent upon all efforts towards improvement, which would be a most unfortunate condition, as the world cannot be allowed to stand still. But the ultimate effect will, however, be found to be the elimination at a much earlier stage than otherwise of such projects as fail to possess a substantial basis of principle, and this would obviously tend to the advantage of such as do possess the necessary qualifications.

Tendering for works.—Works of any important extent are usually submitted to tender, either by public advertisement or private invitation. The obvious objection to the first alternative is the possibility that the work may fall into the hands of one who is utterly incapable of bestowing efficient treatment upon it, in which case it is, in every sense of the word, vain to expect a successful result. To provide against such cases, whenever a contract is advertised, a constant practice of inquiry should be adopted, as to the capabilities of the contractor whose tender appears to be most acceptable. If this is done in a systematic, straightforward manner, no suspicion or resentment will arise, such as would follow any occasional or capricious adoption of the practice. During the prosecution of such inquiries, perfectly satisfactory reasons may be discovered why the

contractor in question is in a position to make a specially good offer for the work. Obviously, in all cases of private invitation, it must of necessity be assumed that this question has received attention in the first instance. It cannot, however, be presumed that the lowest tender, received from invited parties, must necessarily be accepted.

Uniform conditions should apply to all tenders.—In all cases in which work is submitted to tender, one of the most important points to consider throughout, is that each contractor shall be fully informed upon every condition to be observed in the execution of the work. In most cases this condition is incomparably best fulfilled by the provision of a fully detailed and explicit specification, to which each tender must apply, and in which no clause shall be open to two interpretations where one will suffice. But in many cases, and especially when the subject is one with which the contractors are especially familiar, this is not necessary or even desirable. Here considerable variation in the prices submitted for nominally the same work may, very fairly, be expected. This very seldom happens in connection with building work or with ordinary engines, but very frequently in connection with engines and machinery for special purposes. In such cases, before arriving at any decision between the several proposals, it is most desirable that they should be compared, with reference to the following points.

Quantity of material.—Firstly, as to the weight or quantity of material proposed to be supplied. This comparison, however, only applies strictly between materials of the same kind, as it often happens that heavy details may be replaced with advantage by lighter ones of a different, and usually a more costly character. Broadly speaking, however, the utility of a machine is

very intimately connected with the strength and rigidity provided by weight of material.

Quality of material.—Secondly, as to the classes and amounts of superior materials employed, such as gun-metal, phosphor bronze, high-class and hardened steel, etc. The judicious use of these materials often imparts a durable character to the object in question not to be otherwise secured; and anything which is calculated to extend its life usually increases the utility from the very commencement of its use.

Workmanship.—Thirdly, the workmanship bestowed upon the subject. The circularity and finish of bearings, and the truth of flat or lined surfaces, affects very largely, the cost without any tangible or outward signs. But the results of indifferent work at once take effect in great increase in friction, very soon developing into excessive wear, and in various ways militating against the working value of the whole.

Design.—Fourthly, the superiority or otherwise of the design is largely embodied in the points above referred to. But a design which presents a good, outwardly harmonious appearance may usually be taken as evidence of competent attention to points of superiority, which only declare themselves in practical work.

General efficiency.—Fifthly, the whole should be considered with reference to the several points dealt with in the early part of this chapter. Cheap work is not always and necessarily to be avoided. It may be due to special facilities for production, and its adoption may be desirable for reasons already referred to. But the temptations presented by it should only be accepted after the most careful and thorough consideration.

Special services.—The adoption of specialities is a question frequently arising, and in connection with which it may be remarked that the special price put

upon the service which is demanded, often appears excesssve for what, upon the surface, may appear to be only a happy thought. But in many such cases, success depends very largely upon the continued exercise of extraordinary skill, specific experience and attention to detail; the nominal point secured may be quite subsidiary to these.

Articles which show the makers' name are usually to be preferred, and those possessing lengthy unmeaning names should be avoided. Exceptions arise, but they are very few.

CHAPTER L.

MAINTENANCE OF MACHINERY.

Cost of maintenance reduced by reason of convenient access to parts.—As elsewhere repeated, ample means of access should be provided to all parts for examination, cleaning, etc.; in many cases so that inspection may be possible when the machinery is at work. Important work should be arranged with stages, ladders, and hand-rails, ready fixed, so as to avoid loss of time involved in rigging temporary stages, which again are never so safe or satisfactory in use. Good daylight should be provided if possible, but if this cannot be done, the provision of artificial light at all points should be considered. Good means of access frequently lead to the prompt detection of derangement of details, and of interruptions in lubrication.

Cleaning of machinery.—Machinery at all times, and especially when at work, becomes coated with grease and foreign matter, which, for the sake of appearance, ought to be regularly removed. But a more important reason for the operation is connected with the examination of the work, which to some extent is made simultaneously by the men engaged, and usually supplemented by others who are able to effectually inspect

the clean parts. When cleaning is properly performed, any looseness of details is observed, also any brass or metal dust which may arise from defective lubrication. In an engine, the main bearings, the backs of the cranks, the undersides of connecting-rods and eccentrics, the cross-heads, the piston-rods, valve-spindles, and pump-gear should receive the first attention. Then the fly-wheel, governor, lubricators, and all solid bolts should be cleaned, after which the cylinder and valve-chest covers, and all bright work about the slide-bars, connecting-rods, etc., well rubbed down. Lastly, the painted work may be cleaned by greasy waste or sponge cloths. The under or back side of every detail is of importance, at least equal to that of the visible side, and in examining an engine as to cleanliness, the hand should always be passed where the eye cannot reach. By this means, objectionable deposits of hard grease will often be found upon glands, valve-spindles, and pump-rams. Lubricators and drip-receivers should be regularly emptied and cleaned; they should also be frequently searched by passing the finger round the inside, in which no grit or gummy or dirty oil should be found. The teeth of a fly spur-wheel, the surfaces of a belt wheel, and the grooves of a rope wheel, should be kept clean, as in all cases dirt interferes with the proper discharge of the duty, and if such dirt should become dry, it is apt to fall out and cause mischief. In no possible case should the use of emery in any form, or any gritty substance, be used for cleaning purposes. High polish is of much less importance than the removal of tangible deposits, and the cleaning process, though in itself of considerable value, is of more value as evidence that the work receives regular attention and examination. As a rule, stationary engines of the largest sizes receive most commendable

attention, but engines of smaller power are often sadly neglected. A well-managed locomotive, just brought from the shed, may be taken as a model in this respect. High polish is not a leading feature, but every nut, cotter, pin, and other detail shows that it has been cleaned on all sides, and it is impossible to believe that anything can exist in a loose or deranged condition.

Collection of drip.—Drip-pans should be provided to collect every drop of oil, grease, water, or spray, so that floors and walls may be preserved in a clean condition. Small brushes may be used to collect drops from such parts as the cranks and connecting-rod ends, whereby is avoided the dirty scattering of oil which often occurs. Wherever possible, all bearings, or parts lubricated with oil, should be arranged to drain into receivers cast solid with the main castings, frames, etc., of the work. The primary reason for this is the general desire for tidiness. But if grease or water is allowed to soak into foundations, great mischief may be caused in softening and loosening the structure. This is less likely to arise if the whole is well constructed, but this should not be presumed upon, especially in connection with old work. It is also impossible to say that new conditions may not arise by reason of a change in the properties of the lubricant used.

Cleaning of shafting.—Shafting and gearing should receive the same kind of attention as engines. The lubricators are of chief importance, but holding-bolts, keys, and fastenings should be thoroughly well cleaned, also the bosses of wheels, ends of teeth, and the entire surfaces of shafts, bearings, couplings, and pulleys. As in the first case, everything should be so directed as to promote early disclosure of any derangement in the work.

Protection of gearing to exclude dust and solid objects. Wherever possible, and more especially where the atmosphere is highly charged with dust, means should be adopted with the object of excluding the same from free contact with all important bearings and moving parts. Wheels should be cased by strong covers of galvanized iron or blue-steel sheets, arranged to take apart easily for examination of the teeth. These covers should fit as closely as possible, for the exclusion of dust, they should confine all flying splashes of grease, and, above all, prevent accidents in which either the limbs of the attendant, or any tools, such as brushes, ladders, etc., may become entangled. It is not sufficient to provide against probabilities of this kind, but to consider all possibilities whereby a man may stumble or slip, or, having his attention diverted, may unconsciously make a false move, or may be caught by loose clothing, under any conditions of lighting.

Exposure of wheel-keys.—As a rule, the keys of wheels should not be covered out of sight, but in all cases they should be cut off short, and all roughness removed, so that but little risk will remain.

Simplicity of means of protection.—Though proper means of protection should be afforded for the prevention of accidents, these should be secured in the simplest possible manner. In fact, complication in such cases is equivalent to imperfection, owing to the excuse which may appear for the disuse of the appliances, or the postponement of their replacement after removal.

Protection against accidents to persons.—Revolving shafts are usually safe if they are free from projections. The same often applies to belts and ropes. But there is danger in all cases in which two moving parts, or one moving part acting in opposition to an

adjacent standing part, tend to hold, or to draw in, any object which unfortunately approaches too closely to the place in question. Practically, all shafts erected at the present time are turned smooth, and polished all over; when this is done, and all dangerous key-ends, projecting coupling-bolts, and like details avoided, there is comparatively little risk of entanglement of belts round shafting. It is, however, most desirable to provide means for hanging heavy soft belts clear from the shaft when off the pulley; and to avoid running off the belts except when quite necessary.

Prompt attention to defects.—Every defect which may be discovered should receive the earliest possible attention, whether arising from wear and tear, or the slacking of screws, or the slipping of details, keys, or wedge adjustments, or the complete or partial breaking apart of any detail. In all cases delay is likely, and in the majority of cases certain, to lead to an extension of the trouble.

Preservation of correct alignment.—The ordinary wear of important brasses should be watched in its effect upon the correctness of adjustment of the parts; and gauges and gauge-marks made, so that an absolute check may be applied at any time with little trouble. In this way the level of a crank-shaft should be checked. Also the clearance at each end of each cylinder, and the lead of each valve at end of stroke. These are recorded by marks on the piston-rods and valve-spindles, and the use of gauges applied to the solid parts of stuffing-boxes. When such tests are made or repeated, the cylinders should be in a thoroughly heated condition. All main shafts connected by toothed wheels should be noted, so that if any irregularity in the depth of gear should arise, it may be at once ascertained whether wear or the slipping of fastenings has taken place, or

settlement of foundations or walls. For this purpose, light iron rod gauges are useful, each measuring from the side of a shaft to a fixed point, and having a label attached to it, giving its purpose, and its length or other significant dimensions. These should be carefully preserved, and noted in a descriptive list for reference. In belt or rope gearing, the absolute relation of the centres to each other is not of the same vital importance as in tooth connections, but the same means will often prove useful.

Measures to be adopted in cases of minor failure.—When a crack or sign of weakness is noticed in any detail, it may generally be assumed that the part in question has been exposed to excessive strain, which may be repeated, and that though the work appears safe for the time, it should not be allowed to continue in use without the adoption of measures of relief. In any case, the strain to which the part is exposed should be reduced. Any measures which may be possible by way of repair should be adopted at once, to avoid the risk of extended damage and increased cost.

Repair to damaged wheel.—The case of a damaged wheel-tooth may be taken as an instance of the general course to be pursued in repairing operations. When a crack is found in the root of one tooth of a wheel, the whole tooth should be cleaned off for examination, which should also extend to the whole wheel, and especially to several teeth on each side of the damaged one. The examination of the damaged tooth will probably show that it has been exposed to a greater pressure than the rest, and perhaps that the main part of the pressure has been applied nearer to the point of the tooth than in the other teeth. If the crack is a small one, it will be probably considered safe to chip the tooth to a lighter bearing, by using very sharp

chisels. If a crack has extended further, so that the jar caused by chipping involves too much risk, a rough file may be used. The state of contact at such times should be judged by new marks and not old ones, on account of the changes in form due to cracks. In all cases except the very slightest it is advisable to insert soft steel or delta metal screws, or studs through the rim of the wheel, near the centre of the tooth, giving as much strength as possible, without weakening the rim too much by the holes cut through it, which obviously will be almost, if not absolutely in a straight line. By these means, a tooth which has suffered on account of having taken more than its share of work may often

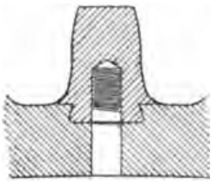


Fig. 153.—Tooth inserted in wheel.

be dealt with so as to prolong indefinitely its useful life. But if the crack is afterwards found to extend, or if the wheels generally show that they are heavily loaded, the tooth should be removed at once, and a false one fitted. This tooth should be made of wrought-iron, steel, or tough bronze. As a rule it should be fitted to the rim of the wheel by dove-tailing, and the base should be spread as far as practicable, so as to give a good effective bearing, especially over the two outer parts, while the middle third of the width should bear more lightly, as in Fig. 153, in which such light bearing is exaggerated to show actual clearance. As many screwed studs should be fitted as the rim will bear.

These studs (as also those inserted into a cracked tooth) may with propriety be placed slightly nearer to the loaded side of the tooth; but this should not be carried so far as to run the risk of jerking out the tooth by backlash. When a wheel is put to work after the fitting of a false tooth, it should be watched, and reduced if necessary, to avoid inflicting upon it more than its fair share of work.

Obviously, the tooth must be kept sufficiently full to perform at least a moderate amount of work, otherwise it exists to no purpose, and probably the next tooth will break. When any tooth in a wheel bears most heavily at or near its point, it is much more likely to break off than if its chief bearing is near to the pitch-line. Consequently, a pair of doubtful wheels which bear near to the points are chipped over. A false tooth in an unshrouded wheel should receive special attention in this respect. In the first instance this must be directed with a view to prevent the overloading of the tooth itself. But anxiety on this point may lead to its being underloaded in the first instance, or in consequence of wear. When this is the case, the adjacent teeth will be correspondingly overloaded. Therefore, special attention should be directed to the four or five teeth on each side of the new one; and any excessive bearing upon these, especially if near to the points, should be at once relieved. In this way several consecutive teeth in a wheel may be fitted to replace broken ones, and the wheel worked for many years with perfect efficiency. In fitting teeth into a shrouded wheel advantage is sometimes taken of the shrouds to such an extent as to impart a strength greater than that of the original tooth of cast-iron.

Application of same principles to other cases.—Similarly, in other cases, damaged parts should be

promptly relieved from strain, repaired, and again put to work, special care being taken to prevent the imposition of excessive stress or shock upon the repaired work, while equal care should be taken to ensure that the repaired work is not allowed to escape duty altogether. At all times, whether repairs be partial or complete, a careful watch should be kept for any repetition of the trouble, and for any cause which may lead to it.

Cracks arising from original strain.—Cracks in cast-iron sometimes arise from irregularities in substance, and consequent strain set up in the casting. The crack itself may develop after many years' use, and may or may not be serious, according to its position and the attendant conditions.

Attrition of substance.—When red dust is found sprinkled over any surface near to an unlubricated, fitted or riveted joint in ironwork, some movement of the surfaces upon each other is taking place. Fine dust of iron is separated by the attrition, and becomes oxidized by the air. When this is observed, no time should be lost in securing the parts, before the fit is quite destroyed. Sometimes, an original defect in fitting is disclosed in this way, sometimes one accidentally developed, and sometimes one caused by inaccuracy of lines. The same phenomenon is occasionally observed in bearings which suffer from gross lack of attention. In some cases there is evidence to show that the film of oxide produced possesses weak lubricating properties.

Hot bearings.—The heating of bearings may be caused by the entry of grit or other matter which causes cutting or roughing of the surfaces, and direct resistance to motion. It may also be caused by loading too heavily for the condition of the surfaces, and the means of lubrication in operation at the time. With ordinary

drop lubrication, it often happens that the flow of oil is obstructed, and moderate, or even serious, heating occurs, but it is very seldom that the supply is suddenly and totally stopped. The means now adopted in good practice, whereby oil is poured on to a heavy bearing in a copious stream, are practically perfect, with most reasonable attention. But when neglected, the supply is liable to become stopped, when heating develops very suddenly and rapidly.

Distortion of brasses by heat.—In all cases of heating of shafts, but especially such as last described, the surfaces of the brass and of the shaft reach a temperature much higher than that of the metal in the body of the shaft, or nearer to the outside of the brass. This condition causes a brass of ordinary pattern to expand more at the inner part than along the outer part, and to tend to flatten out to a greater radius than before. But the solid block in which the brass is bedded prevents this expansion, in which operation a strain is inflicted upon the brass, precisely as though the cold brass were bent inwards by an amount equal to that which it would have been bent—or unbent—outwards by free expansion. When the heated brass afterwards cools down, it contracts by an equal amount unless forcibly restrained, and closes upon the shaft with an amount of force due to the undeveloped contraction. This force due to undeveloped contraction may cause the brass to break apart or to crack all over the bearing surface. If the brass does not break at once, the force by which it contracts upon the shaft will cause renewed heating, and a repetition of the same conditions, until the brass is refitted or is broken. Similarly, the brass may be broken in its efforts to expand. The conditions above described apply in the simplest form to brasses whose thickness is uniform throughout the circumfer-

ence, but brasses of other proportions are similarly affected. Great thickness, if uniform, is an advantage; but in connection with smaller thicknesses is likely to cause fracture through the weaker parts.

Straining of shaft by heat.—The strains which act upon the material of a shaft are of the same character as those which act upon the brass, the material along the surface tending to expand more than that below. The shaft, being of a solid circular section, cannot yield in any way comparable to the brass—hollow shafts of any proportion usually adopted are in this respect practically equal to solid shafts. The consequence is that the material at the surface is under pressure, or in an incipiently folded condition. The contraction on cooling then causes fine surface cracks in the material, of which the longitudinal ones are the most apparent. A hollow shaft will cool more rapidly than a solid shaft, and this increases the difference between the temperatures at the surface and in the interior. When a hollow shaft becomes heated, each end should be stopped tightly with a piece of waste, and on no account should water be applied for cooling the interior.

Damage to shaft caused by heating.—There is no reason to suppose that the strength of a shaft is seriously affected by the cracks which are caused by one instance of accidental heating. But if the heating is repeated a few times, the cracks extend, and short ones break into each other. This is increased by the flexibility of the shaft; and few shafts are so strong as to be safe after heating a few times, sufficiently to cause the development and extension of surface cracks. No information is available to show how this condition of the surface affects the frictional resistance, but there must be some direct loss in this respect. Shafts have been broken in cases in which there is evidence to show that

the fracture has originated in strain set up by rapid heating and cooling. A shaft which has been once subject to such misuse should therefore always be attended by every precaution which can reduce the liability to renewed heating. In the absence of surface marks there is, however, no evidence to show that a shaft has sustained any injury in consequence of the heating.

In all cases longitudinal cracks in a shaft are of much less importance than circumferential or oblique cracks. Many shafts are working in absolute safety notwithstanding the existence of cracks of very considerable magnitude, and which would be absolutely fatal if they ran in any direction other than a strictly longitudinal one. But in all cases where a crack is found, it should be noted in such a way as to show at every opportunity whether any extension takes place. The cracks should be filled with oil, and bending and torsional force applied, when any flexure of the shaft will be shown by movement of the oil.

CHAPTER LI.

JOINT MAKING.

Jointing surfaces.—Pressure-tight joints are required for making good the edges of covers and other parts which are required to be alternately opened for internal cleaning and examination, and afterwards closed. If the surfaces in contact are carefully fitted together, tight contact may be made, metal to metal, or if the surfaces are a little less perfect, a smearing of boiled linseed oil, or even tallow, will be required. The surfaces may be worked together by scraping or grinding, or both, and all grit should be carefully excluded in bolting the surfaces together. Sometimes, however, notwithstanding precautions, some little foreign matter may become interposed between the surfaces, or some change may arise, either from varying temperature, or from inherent strain or other means, which interferes with the perfection of fit existing between the surfaces, so that an absolutely tight joint becomes difficult or impossible of attainment, and leakage results. With a view to avoid this difficulty, a packing of some material softer than the main surfaces may be interposed between the two, which may be

compressed so as to tightly fill the space, and prevent leakage, notwithstanding slightly adverse conditions.

Materials used for jointing.—Joints under pressure have been packed with yarn of flax, hemp, and cotton, sheet mill-board or paper, putty of red and white lead, brass wire-gauze covered with putty, "Tuck's packing," which consists of canvas coiled round itself, and made up with different cements, with or without an india-rubber core; also combinations of canvas and india-rubber made up in sheets, india-rubber alone, gutta-percha cord, lead and copper wire, lead pipe, flat copper, thin corrugated metal, and asbestos in its different forms.

Conditions of application of different packings.—With ordinary pressures, when the surfaces are in good condition, the very least thickness of packing suffices to make the joint tight; and it will remain tight, unless some change arises in the packing. But if the packing should be unable to resist decomposition, under the conditions to which it is exposed, a leakage may develop. When this happens, there is every reason to expect it to increase, by reason of the continued wasting of the packing, and, what is much more serious, the corrosion or mechanical disintegration of the surface by the rapid cutting action of the escaping fluid.

Joints made with cotton yarn upon good surfaces are at first quite tight under high-pressure steam, but the yarn very soon burns out, when a slight jar, or a strain by cooling, or other cause, will start leakage, which can only be stopped by taking apart, cleaning, and re-making the joint, which should be done at the earliest possible moment. Flax and hemp yarn, and mill-board, as applied to joints, possess very much more substance than cotton yarn, and do not suffer so soon -- -- extensively from burning; thin paper is also a

good material for joints. Putty of red and white lead, with a little boiled linseed oil is also excellent. If the surfaces are in a defective condition, a thickness of fine brass wire-gauze will often suffice to make a tight joint with the putty spread on both faces. If the surfaces are in too bad a condition for this, a second thickness of gauze may be applied with putty between the two. If this should prove to be inadequate, the surfaces must be in such a condition as to call for re-facing, if this can possibly be done. Sometimes it is found to be impracticable to re-face a joint, either on account of its inaccessibility, or deficiency in thickness of material for strength, or for other reasons, when it will probably be necessary to use Tuck's packing. Lead wire is a convenient packing to use at moderate temperatures; it is, however, found to change its position by reason of alternate expansion and contraction by changes of temperature, so that there is a limit to its endurance in an efficient condition. Copper wire and flat copper being harder, bed themselves slightly into the surfaces, and are therefore less likely to creep out of position, but are less elastic than soft packings. Lead pipe was formerly much used for man-holes and mud-holes of boilers working at low pressures, but the necessity for a plumber's joint of substance and strength uniform with that of the pipe itself is the chief difficulty with the plan. The use of thin corrugated metal rings avoids the trouble involved in cutting out rings from wire gauze, but they do not hold the cement or putty so well as gauze. Asbestos putty is used in precisely the same way as red-lead putty, with or without gauze. For thicker joints, sheet asbestos may be cut to shape, or purposely made rings may be used; these answer excellently for boiler joints, but when adopted in engine-work oil is apt to percolate through the mass of

asbestos. Sheet india-rubber compositions with canvas insertions have been much used, but owing to the tendency to decomposition by exposure to heat—by which they assume a brittle condition—they are now comparatively little used. Rings of india-rubber, with or without canvas insertion, are often used for work not exposed to heat, such as water-pipes connected by flanges. Their elasticity, especially when free from canvas, is often of great use in work of this kind. Ordinarily, they may be repeatedly used. Gutta-percha cord is used only for hydraulic joints, in which it is secured in joints of a dovetail section. It cannot be used with safety at temperatures above 75° F., owing to its liability to melt. Leather and vulcanized fibre are sometimes used for joints exposed to water at great pressure and low temperature.

Bolting of joints.—In connection with the majority of joints exposed to pressure, and made tight by a packing, the latter is kept in its place by reason of pressure applied by bolts. The most suitable distances between the bolts depend upon the rigidity of the several pieces. The bolts must possess sufficient strength to apply the required force of compression upon the packing, and usually to support the internal pressure upon the cover in addition. If the surfaces were absolutely true and rigid, and the packing of uniform thickness throughout, the width of packing, within rather wide limits, would be immaterial. In practice all these conditions fail more or less completely. Consequently, some portions of the packing must suffer excessive compression, or some other portions will fail to receive sufficient pressure to prevent it from being blown out on the application of pressure. Similarly, the same failing in a less degree causes leakage. When leakage arises in such a way, there is every probability that the surfaces and the

packing will suffer in consequence, and the blow-out will only be postponed for a time. Surfaces which have been so long in use as to have suffered much wasting are often exposed to great risk of this kind. In most cases, this difficulty is efficiently met by keeping the packing rather narrow, and of thickness sufficient to allow for elasticity of compression. When the force of the bolts is applied to a packing only 1 inch wide, it is obvious that the resulting compression will be three times as great as it would have been if the width of the packing had been 3 inches; and obviously the amount of irregularity which is sufficient to prevent the secure grip of the packing is, in the latter case, only about

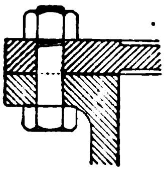


Fig. 154.—Bolted joint to resist fluid pressure, wide facing.

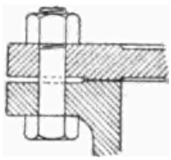


Fig. 155.—Bolted joint to resist fluid pressure, narrow facing.

one-third of that in the former. In good work, the usual practice is to face the surface over with the entire width of the joint, and to cover the whole with jointing material, as in Fig. 154, before screwing down. This usually makes a good joint, quite tight, and of excellent appearance, but not well adapted for application to defective surfaces. Formerly, the fitting surfaces were left narrower, as in Fig. 155, so that packing of moderate elasticity could be screwed down so far as to hold the packing tightly all over, and thereby to make a more secure joint than the first, with the same tension upon the bolts and strain upon the flanges, though leaving an opening along the edge which is sometimes considered unsightly. But the same advantages may be

secured by confining the application of the jointing material to an equal width of the straight flanges in Fig. 154, or still better by reducing the step in the surfaces to very small dimensions. In ordinary cases the packing should be made 1 to 1¼ inches wide, and grooves may be turned or cut in the surfaces. Obviously, narrow surfaces require the jointing material to be applied to them with accuracy over the whole surface.

Thickness of packing.—Though the thickness of the packing must in all cases be sufficient to provide elastic compression to cover irregularities of surface, yet it should not exceed this, on account of the increase of area upon which the internal pressure may act to blow out the packing, and also because the appearance would probably be objected to.

Softening of packing by heat.—Tuck's packings are often hard when cold, and soft when hot; they should therefore be screwed up carefully and repeatedly as the temperature rises, or they may be blown out. When this kind of joint is used for boiler mud-holes or man-holes, a lip of some kind should be prepared to prevent with absolute certainty any blowing out of the packing. In such joints, if a thin packing is used from the first a good joint may be always kept, but if allowed to leak the surfaces will soon become indented and thick packing will be always necessary to make a tight joint, unless the surfaces can be re-faced, so that a very small neglected leakage often leads to long-continued trouble.

Repeated use of packing.—When asbestos or similar material is used, one side of the packing, or one metal surface may be dusted with black-lead or French chalk, which will prevent adherence upon that side, though the tightness of the joint will be unaffected. By this

means, the packing will remain attached to the other surface next time the joint is parted, and thus the packing may be efficiently used several times. But unless the whole of the packing can be used unimpaired, it should be discarded, and the surfaces scraped clean, especially against bolts and studs, where small pieces are apt to escape observation and give trouble.

Use of through-bolts.—When bolt-heads are exposed to pressure, and the points are open to the atmosphere,

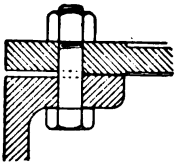


Fig. 156.—Narrow joint, with head of bolt inside vessel.

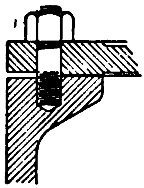


Fig. 157.—Narrow joint with stud.

there is a possibility that leakage along the bolts may become developed. This is prevented by a grummet of yarn against the head of each bolt, made tight by putty. In such cases, the packing may be made of the full width of the flanges, but the arrangement shown in Fig. 156 is better. When studs are adopted they should be arranged as in Fig. 157, and not placed in holes drilled quite through the metal, so that their points would be exposed to steam.

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